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SOME ASPECTS OF PROPELLER EXCITED VIBRATION

by

T. A. LAMPLOUGH

LLOYD'S REGISTER OF SHIPPING
71, Fenchurch Street, LONDON, E.C.3

Some Aspects of Propeller Excited Vibration

By T. A. Lamplough

Ship vibrations of one sort or another have troubled most Naval Architects and presented them with problems that are sometimes difficult to overcome.

Vibrations of the main hull, either engine or propeller excited, and occasionally hull vibrations excited by the ship driving into a head sea, are all possible. Each type presents its own set of problems for solution.

Another type of vibration, which occurs and which also emanates from propeller excitation, is of a local nature. Generally the worst effects are felt in the region of the sternframe and hull at the after end of the ship, in the immediate vicinity of the propeller. The trouble manifests itself in the form of forced vibrations of the surrounding structure.

The forces causing the vibration are of two types:—

(1) Those arising from the pressure field set up in the water surrounding the propeller by virtue of its action and transmitted to the adjacent structure, which are called "surface forces".

(2) Those transmitted through the propeller shaft bearings into the hull structure, which are due to the unequal thrust and torque variations from the propeller, and designated "bearing forces".

It will be clear that these fluctuating forces bear some relationship to the number of propeller blades, and the shaft R.P.M. In fact, their primary frequency is N $\times$ R.P.M. cycles per minute, N being the number of blades on the propeller.

The adjacent hull structure, plating panels, rudder, etc., are all subject to forced vibrations, and if the natural frequencies of the various parts of the structure are coincident with the forcing frequency, then resonance will occur, and trouble may arise depending on the violence of the effect.

Owing to the complexity of a ship's structure, particularly at the after end, it is not usually possible to prevent the problem arising by ensuring that the natural frequencies of individual items are not coincident with the forcing frequency. The method of solution is to attempt to limit the exciting forces to an acceptable level. By the latter is meant the acceptable level from the viewpoints of structural damage and crew comfort.

It will be of interest to look generally at the reason for this thrust and torque variation, and to have some idea of the magnitude of the forces and moments available for excitation.

Propeller Action

Simply stated the action of a marine propeller is to take water travelling at a velocity A at the propeller disk, and accelerate it astern of this position to a velocity B, where B $>$ A, thereby creating a difference in momentum and an accompanying force, the reaction of which thrusts the ship forward.

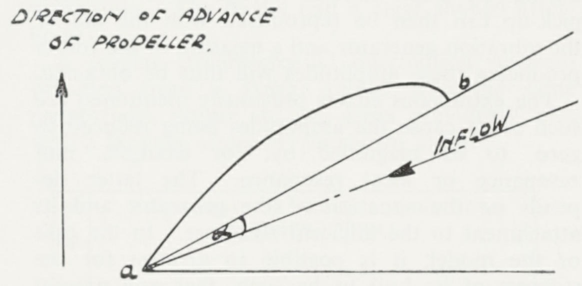


Fig. 1

If we look at a section of a propeller blade (fig. 1), we see that the water enters the blade at some angle α to the line a — b. This angle is called the angle of attack. If the propeller is to produce constant thrust and torque, it is of prime importance that this angle of attack is constant. This condition of constant angle of attack is not attained in practice, the reason being, that the water which makes up the wake, is composed of streamlines of varying velocity. The effect of these variations over the circumference of the propeller is to cause the angle of attack to vary. It is this periodical variation which is the principal cause of thrust and torque fluctuation.

Forces and Moments available for excitation.

Measurements of propeller forces and moments causing vibration of the type under consideration, are available from two sources:—

(1) Model experiments,

(2) Actual measurements taken on ships in service.

The measurements obtained from both these sources must be regarded in the qualitative and not the absolute sense, the reason being the difficulties involved in measuring only propeller forces and moments, without any extraneous effects being included in the results being recorded.

The method of measuring these forces and moments is somewhat similar on both model and actual vessel. For the model, it consists of arranging a means of excitation in the form of a vibration generator, mounted in the plane of the

propeller. The propeller is driven by a motor which also drives the vibration generator. With this arrangement it is possible when running the model, to adjust the amount of the exciting force and also to adjust the phase of this force with respect to the propeller. A vibration pick-up is fitted at the fore end.

The model is run and the amplitudes of the propeller excited vibrations are recorded on the pick-up. The vibration generator is then adjusted and phased so that the maximum amplitudes registered on the pick-up are brought to zero. In this way the forces excited by the propeller on the model can be found.

The forces on the ship can be measured similarly by mounting a vibration generator at the aft end. The maximum amplitudes measured during the ship's normal voyage by a vibration pick-up can then be reproduced with the aid of the vibration generator and a measure of the forces producing these amplitudes will thus be obtained.

The extraneous effects previously mentioned are such as to cause the amplitudes being reduced to zero, to be magnified by, for example, hull resonance or local resonance. The latter depends on the situation of the generator and its attachment to the adjacent structure. In the case of the model, it is possible to arrange for the stiffness of its hull to be such, that any natural hull frequency is well outside the range of propeller exciting frequencies. However, it is not claimed that actual forces that are likely to be experienced on the ship can be predicted.

In reference 1 are some values for propeller forces causing vertical vibration, which were measured on both a model and a full-size vessel of the Mariner type. The propeller was four-bladed, and at 106 R.P.M. on both ship and model, a vertical force of 5·5 tons, which represented some 6·5 per cent of the available thrust, was measured.

On another Mariner type vessel a vertical force of about 7 per cent of mean thrust was measured.

Reference 2. T. W. Bunyan produced the following diagram, fig. 2, for two large tankers, one fitted with a three-bladed propeller, the other, four-bladed. It can be seen that in the loaded condition, for the three-bladed propeller running at 100 R.P.M. the available force is greater than 16 tons which represents some 23 per cent of the total thrust. For the four-bladed propeller at 100 R.P.M. the force is about 8·5 tons, 12 per cent of the available thrust. He also states that torque variations of 12—15 per cent have been measured.

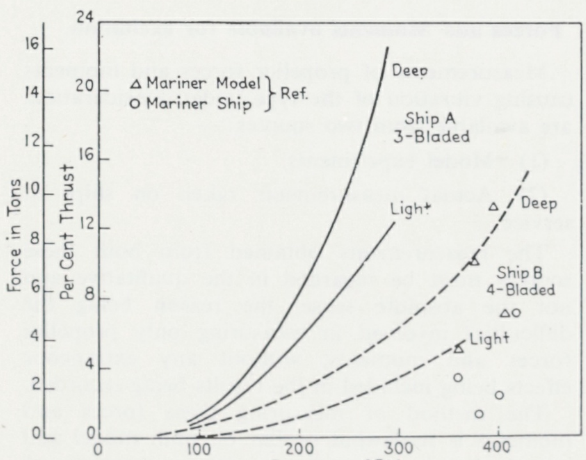


Fig. 2

The following table is reproduced from reference 3. It has a theoretical basis, and gives estimates for percentage of total thrust and torque fluctuations available for excitation for 3, 4, and 5 bladed propellers: —

	Number of Propeller Blades		
	3	4	5
Thrust %	20·2	25·0	4·8
Torque %	12·7	17·0	4·0

Table 1

Remembering that these vibratory forces are reaching maximum and minimum values N times per revolution of the propeller shaft (N is as previously defined), it is not surprising that in the worst cases of vibration of this type, structural damage has been found.

Instances of such damage found are side shell frames fractured above the propeller aperture, after peak bulkhead plating fractured, and general damage in the region of the sternframe and after peak.

In reference 12 the author mentions the case of a cargo ship which suffered from local vibration which, to use his own words, "produced very rapidly a practical demolition of the after body".

The inherent stresses in the fractured members were probably low. However, it must be remembered that the acceptable level of vibratory loading is less than that for static loading. In this connection a value given in reference 4, for the permissible stress acceptable in a structure constructed of normal type steel, which is subjected to vibratory loading, is about one third of the value that would be acceptable for the same structure under a static system of loading.

Wake Variation

This is the principal cause of the type of vibration under discussion. The shape of the vessel's after body, particularly the sections immediately forward of the propeller, will have much to do with the final form of the wake distribution.

When the fore and aft position of the propeller is decided, it is necessary to know the value of the wake velocities at this position, to enable the propeller to be correctly designed. Either, estimates of the strength of the wake must be made from accumulated data, or, model tests must be carried out and wake diagrams produced.

A typical wake survey is as shown in fig. 3, (reference 5).

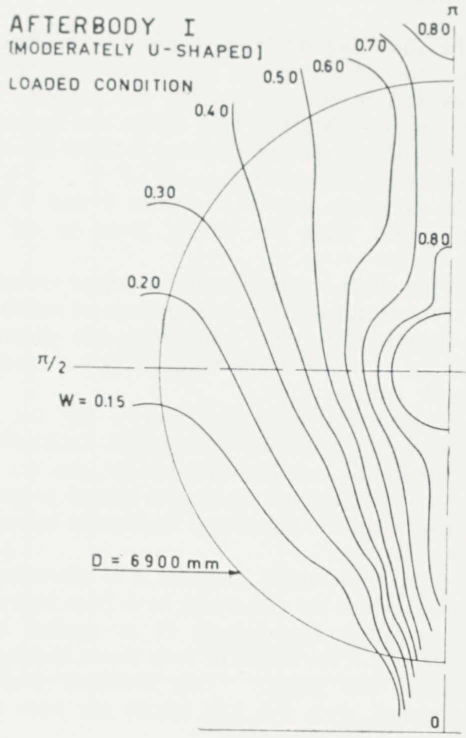


Fig. 3

This is for a 39,000 ton deadweight tanker model, with moderately U-shaped sections aft. The model was tested in the loaded condition, at a speed equivalent to 16 knots on the actual ship.

If we now look at fig. 4, which is obtained from fig. 3, and shows the circumferential wake distribution,

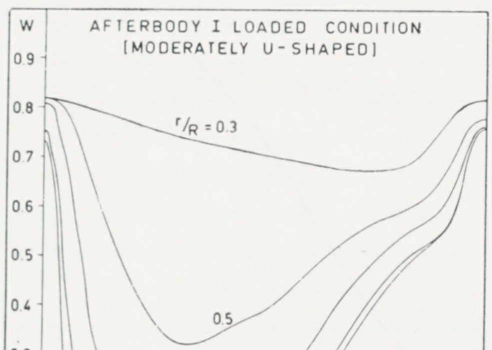


Fig. 4

and consider a point on a propeller blade at ·7R from the centre of the boss, we see that the blade at this point, while turning through 0 — 180°, will be dealing with velocities varying from ·81 W, through ·2 W, to ·75 W, where W is the wake speed expressed as a fraction of the ship speed.

The propeller can be designed to take account of the radial variation of the wake, by giving it a suitable pitch distribution. Such a propeller is known as "wake adapted". However, nothing can be done to allow for the circumferential wake fluctuation. If it were possible to arrange for the nominal centre of thrust of the propeller blade to be at ·3 R, then little fluctuation would arise from this source. An average value of the position of the thrust centre on a blade is ·65 R — ·7 R, and hence the maximum thrust is available in one of the worst positions of circumferential fluctuation.

In the early 30's E. Hogner tested a model with an afterbody designed, amongst other things, to give a minimum amount of circumferential wake fluctuation. The model had a cigar shaped afterbody.

A similar circumferential distribution diagram obtained from a tanker model with the same proportions, power etc., as given previously, and with a Hogner afterbody, is shown in fig. 5.

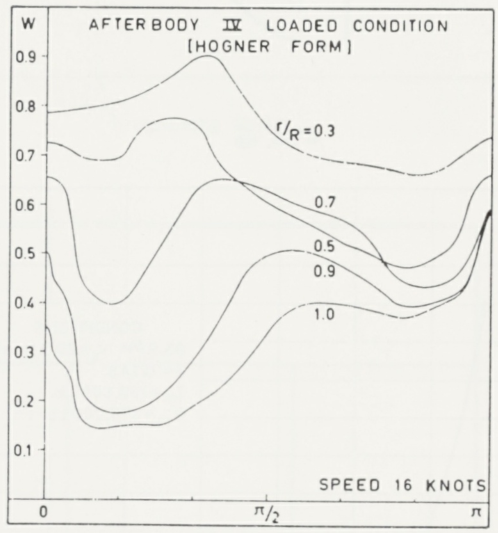


Fig. 5

If we compare this with the previous wake diagram it is apparent that the order of the circumferential variation is less in the latter.

A tanker of 48,000 tons deadweight, 19,000 SHP, with a speed of 17 knots, built by AG Weser, has an afterbody which resembles somewhat the Hogner design. The design was developed after a series of tank tests and attention was focused strongly on obtaining a satisfactory wake distribution in view of the high power involved on the single screw.

From measurements taken on trial, at maximum output, some $\pm$ 4·5 per cent of the mean thrust, about 12 tons of thrust variation, was recorded. Torque variations were found to be negligible. These figures were considered to be sufficiently low.

Propeller clearances.

The clearances between the propeller and the sternframe are important factors to be considered when designing the ship to minimize vibration.

Various authorities give figures for the clearances considered necessary and they have been summarized in a table in reference 11. Incorporating these figures into the design does not automatically preclude the possibility of propeller excitation.

The most important clearances are at points a, b and c. The clearances a and b directly affect the value of the surface forces, while that at c governs whether or not the rudder will sustain any large excitation.

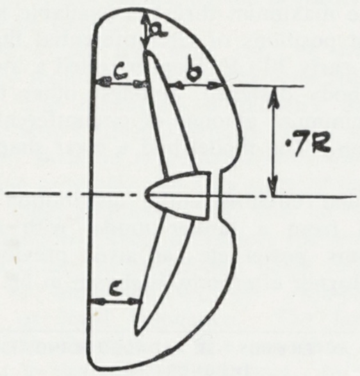


FIG. 6

The following diagram, figure 7 (reference 1), shows the effect on the transverse and vertical forces and torque, of moving the propeller aft. This was for a Mariner type design and the model had no rudder. It was intended to show the effect of increasing the aperture.

The diagrams shown in figures 8 and 9 have been drawn from test results given in ref. 13. They show: —

(a) the effect on the thrust and torque variations for a conventional stern arrangement of increasing the clearances forward and above the propeller,

(b) the same variations measured on a modified Mariner type stern.

The diagrams have been drawn for the first harmonic of the thrust and torque variations in the case of the four-bladed propeller, and the first and second harmonics of the five-bladed propeller, as these were the significant harmonics measured during the tests.

For the four-bladed propeller it is interesting to note that whilst the clearances have been improved upon in stern arrangement II as against stern arrangement I, the values of both thrust and torque variations are higher. The modified Mariner arrangement gives the best results for both four and five-bladed propellers.

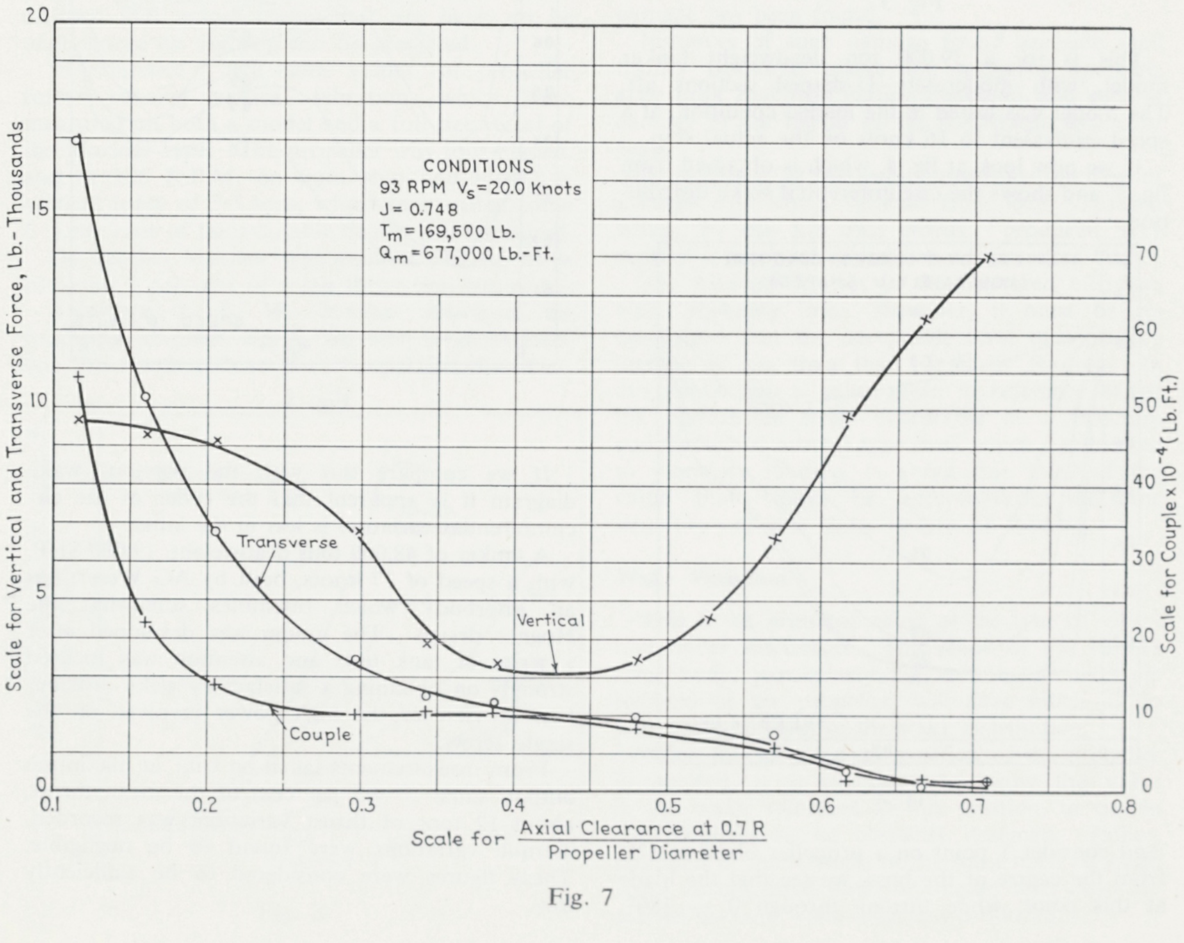


Fig. 7

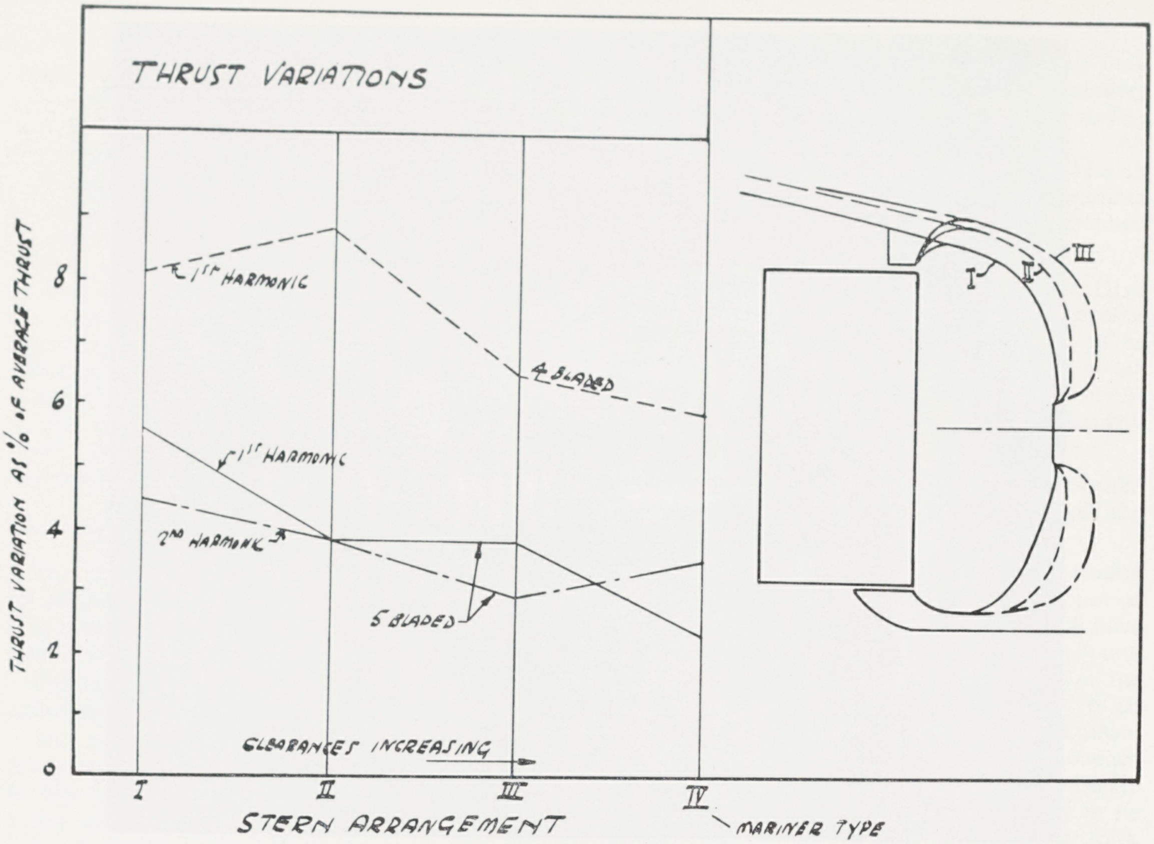

THRUST VARIATIONS
THRUST VARIATION AS % OF AVERAGE THRUST
8
6
4
2
0
1st HARMONIC
4 BLADED
1st HARMONIC
2nd HARMONIC
5 BLADED
CLEARANCES INCREASING
I
II
III
IV
STERN ARRANGEMENT
MARINER TYPE

Fig. 8

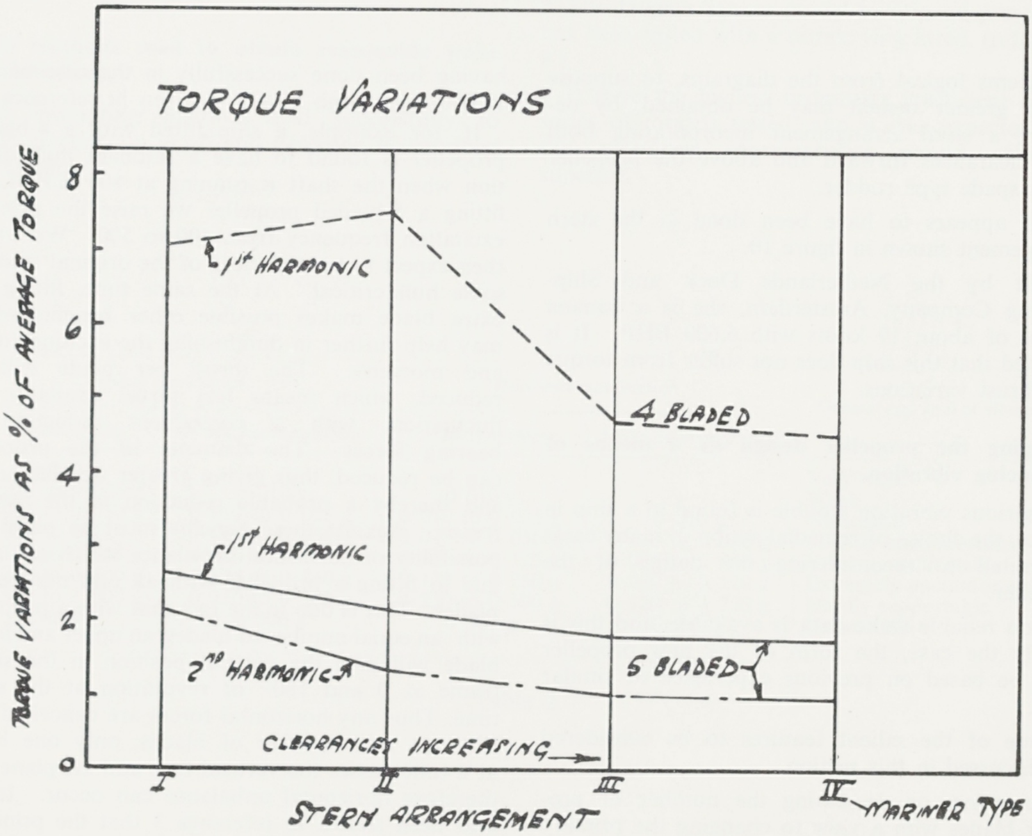
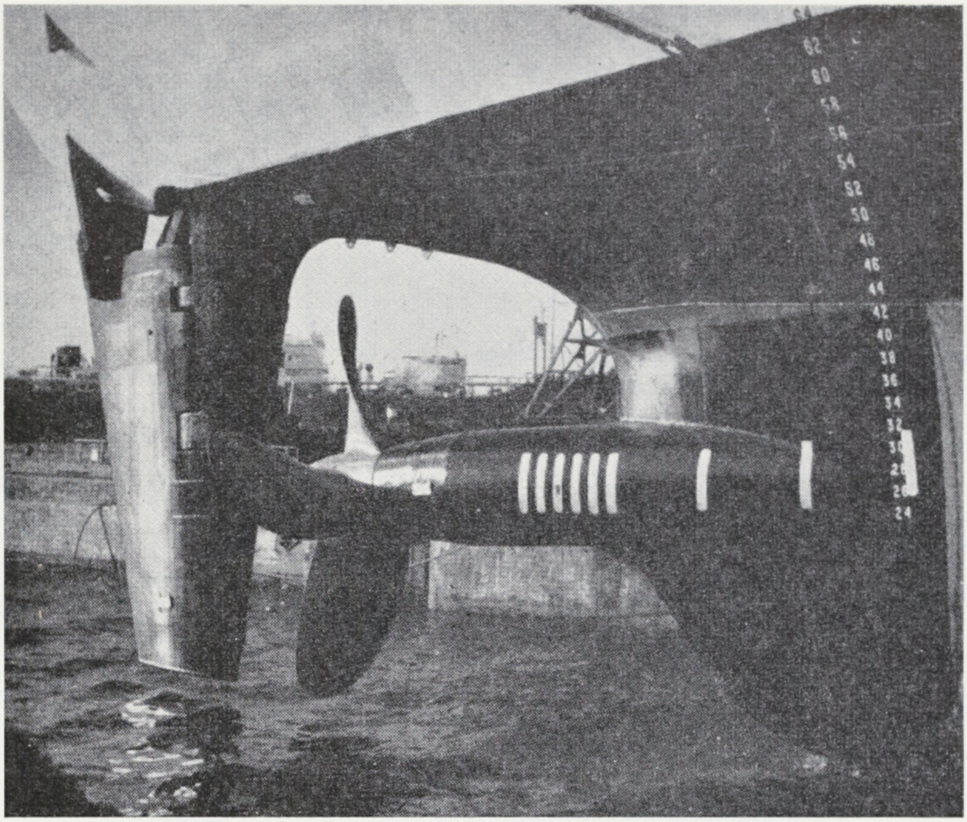

TORQUE VARIATIONS
TORQUE VARIATIONS AS % OF AVERAGE TORQUE
8
6
4
2
0
1st HARMONIC
4 BLADED
1st HARMONIC
2nd HARMONIC
5 BLADED
CLEARANCES INCREASING
I
II
III
IV
STERN ARRANGEMENT
MARINER TYPE

Fig. 9



(*Courtesy of "Shipbuilding & Shipping Record"*).

Fig. 10

It seems logical from the diagrams, to suppose that a greater benefit may be obtained, by designing a stern arrangement incorporating both large clearances forward and above the propeller and a spade type rudder.

This appears to have been done in the stern arrangement shown in figure 10.

Built by the Netherlands Dock and Shipbuilding Company, Amsterdam, she is a banana carrier of about 19 knots with 6,600 BHP. It is reported that this ship does not suffer from torque and thrust variations.

Changing the propeller design as a means of reducing vibration.

If serious vibration trouble is found in a ship in service, the choice of remedial action in many cases is limited to reconsidering the design of the propeller.

If no reliable wake data is available, and this is usually the case, the form of the new propeller must be based on previous experience of similar cases.

Some of the salient features to be considered are discussed in this section.

Increasing, or decreasing the number of propeller blades with a view to changing the primary frequency of excitation. One example of this having been done successfully in the case of two 10,000 tons cargo vessels is given in reference 10.

If, for example, a ship fitted with a 4-bladed propeller is found to have a resonant hull vibration when the shaft is running at 100 R.P.M., by fitting a 5-bladed propeller we raise the primary excitation frequency from 400 to 500. We would then expect to be well clear of the original troublesome hull critical. At the same time, fitting the extra blade makes possible other benefits which may help further in diminishing the exciting forces and moments. The thrust per blade will be reduced, which means less thrust available for fluctuation, with a consequent reduction in bearing forces. The diameter of the propeller can be reduced, thus giving greater tip clearances, and thereby a probable reduction in the surface forces. Against these benefits must be offset the possibility of horizontal unbalance which can arise due to fitting a propeller with an odd number of blades. This is due to the fact that with a propeller with an equal number of blades, an upper and lower blade will pass the vertical position in the sternframe at 0 and 180° of revolution at the same time. Thus any horizontal forces are cancelled out. With an odd number of blades, only one blade at a time passes the vertical fore and aft plane and therefore horizontal unbalance can occur. It has also been shown in reference 3 that the principal frequency of a 5-bladed propeller is that occurring

at 10N. Therefore, if the propeller is being re-designed in an effort to clear local vibrations this can mean that fitting an extra blade may have an adverse effect on the structure at the new frequency.

Changing the pitch distribution with a view to reducing the exciting forces may be of help. In normal merchant ship propellers the highest loading on the blade is experienced in the region of the tips, and it is this region which suffers the worst from the wake inequality. It might therefore prove beneficial to reduce the pitch distribution in this region, and thus reduce the thrust available. Altering the blade shape may also help in this direction. The propeller rake can sometimes be changed to give greater aperture clearances.

Crew comfort.

The tendency in tankers and bulk carriers for all personnel to be housed at the after end of the ship raises the need to minimize this type of vibration from the crew comfort aspect.

Constant high frequency vibration with the associated noise level is very tiresome.

The method of classifying the vibration into levels depending on the acceleration accompanying it, has been used in order to produce the table below, (reference 12).

The last column is an interesting attempt to give the values of the acceleration a more practical meaning.

Novel methods used to obtain reasonable wake distributions.

An interesting method used successfully to obtain a reduction in aft end vibration is described in reference 7.

The trouble arose in several Great Lakes bulk carriers that had their original machinery installation replaced by new units providing approximately 50 per cent greater power. All these vessels suffered from fantail vibration of such severity that the authors stated "it required immediate consideration to permit continued operation".

During model tests it was noticed that a large downdraught of water forward of the propeller caused the water in the region of the disc to be agitated. It was decided to fit fins forward of the aperture and extending over the propeller itself, in order to attempt to eliminate this downdraught. (Fig. 11).

On trials with the fins fitted the measured amplitudes of vibration were one-fifth of the original figures.

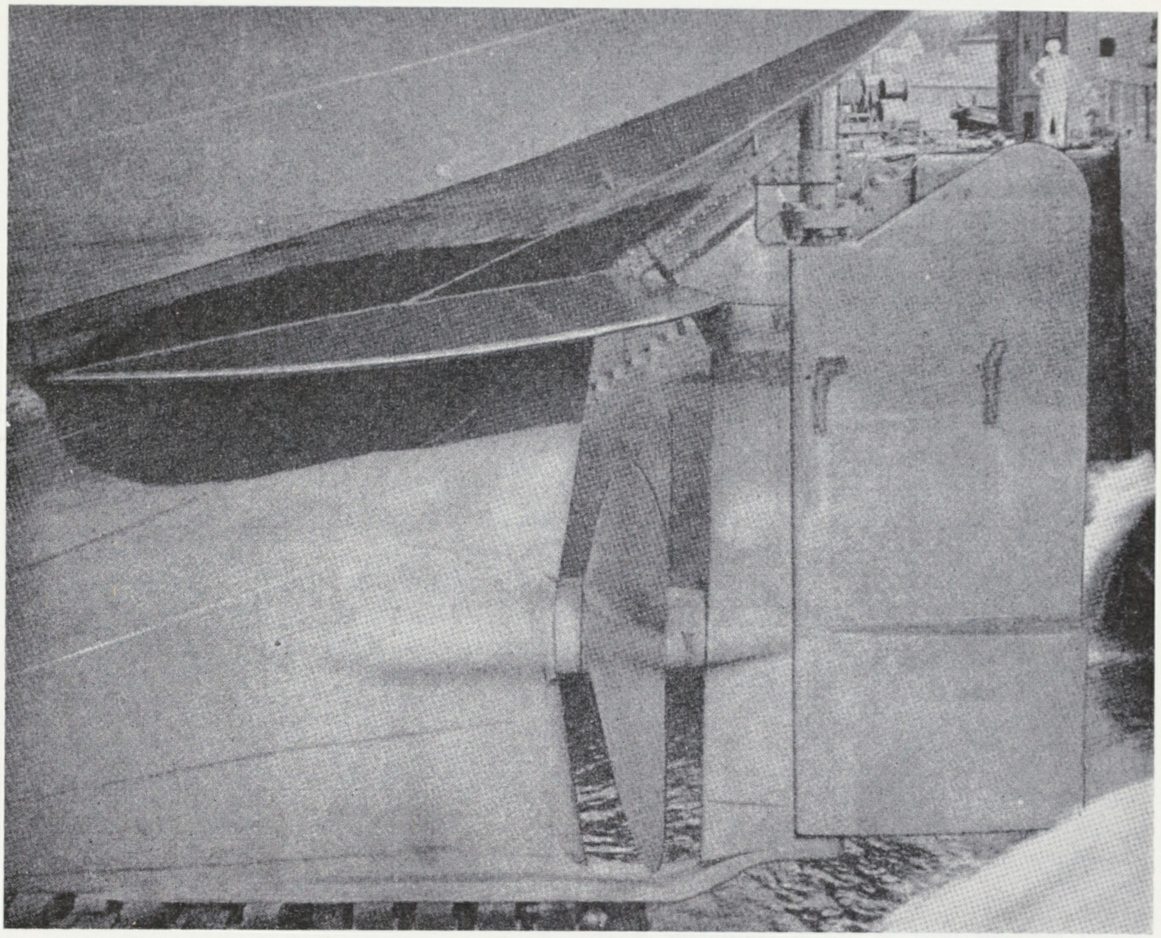
Another method which offers a possible solution is to fit a nozzle round the propeller. The use of nozzles is normally confined to vessels which have high propeller loadings, e.g. tugs, when it is found that fitting a nozzle leads to an increase in the propulsive efficiency. It has not been fitted previously on an actual ship as far as is known, as a means of eliminating vibration. In reference 9 it is stated that it is advisable to investigate the possibility of improvements in efficiency to be obtained by fitting a nozzle arrangement to a tanker of 30,000 tons deadweight, with a speed of 17 knots.

The Hogner tanker model mentioned previously has been tested with a nozzle ring fitted, (reference 8.)

It gave the highest propulsive coefficient of the series of models tested, and it was expected that the vibratory characteristics would also be superior.

Vertical acceleration		Horizontal acceleration		General appraisal of vibration
At ends of ship	In the accommodation	At ends of ship	In the accommodation	
0·010 g				Very weak
0·010 to 0·025 g	0·010 g	0·010 g		Weak
0·025 to 0·050 g	0·010 to 0·025 g	0·010 to 0·025 g	0·010 g	Noticeable
0·050 to 0·120 g	0·025 to 0·050 g	0·025 to 0·050 g	0·010 to 0·025 g	Slightly uncomfortable
0·120 to 0·250 g	0·050 to 0·125 g	0·050 to 0·125 g	0·025 to 0·050 g	Very uncomfortable
0·250 to 0·500 g	0·125 to 0·250 g	0·125 to 0·250 g	0·050 to 0·120 g	Extremely uncomfortable
0·500 to 1·000 g	0·250 to 0·500 g	0·250 to 0·500 g	0·120 to 0·250 g	Hardly supportable
1·000 g	0·500 g	0·500 g	0·250 g	Unbearable

Table 2



Fin fitted to the *Carl D. Bradley*

Fig. 11

Conclusions.

In conclusion, if any doubt exists that trouble of this type may arise in a design, then the model should be tank tested and attempts made to homogenise the wake distribution. In the case of the A.G. Weser tanker *Hadrian,* this gave a satisfactory solution. Failing this, the aperture clearances, particularly that forward of the propeller, should be as great as possible.

The cost of tank-testing a 40,000 ton tanker model with the above in view would be about one sixth of the cost of docking the ship and replacing the propeller if trouble should be found when the vessel is in service. This figure does not include the loss of earnings whilst the ship is idle.

It may be that the ultimate solution will be some stern arrangement such as the Hogner type, plus a propeller fitted with a nozzle.

Much research is being done in national testing tanks on this problem, mainly with a view to giving quantitative data for use in the design stage, to reduce the likelihood of trouble.

The problem, as is usual with many ship problems, presents conflicting requirements that call for compromise. One of the balances to be struck is that between the characteristics of the propeller and its position from the best efficiency viewpoint, and the conflicting requirements from the aspect of vibration.

It must not be inferred from the foregoing that the majority of ships suffer from serious propeller excitation. What is important is that in the last decade, average ship service speeds have increased, with associated increases in powers. These powers are in the main still delivered through single screws, therefore if excitation does occur, the potential is much greater than previously, hence the need to be fully aware of the problem.

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Session 1961-62

Paper No. 1

Discussion

on

Mr. T. A. Lamplough's Paper

SOME ASPECTS OF PROPELLER EXCITED VIBRATION

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

Discussion on
Mr. T. A. Lamplough's Paper

Some Aspects of Propeller Excited Vibration

Mr. J. B. DAVIES

This is the second paper on vibration which has been read to this Association within 12 months, and this does but reflect the increasing interest which shipbuilders and engineers are having to take in this problem. As I remarked when discussing Mr. Hinson's paper last year, it is difficult to know whether more ships are experiencing vibration troubles or whether it is only during fairly recent years that any attempt has been made to find a solution to the troubles.

Mr. Lamplough has confined his paper to one aspect of the general subject of hull vibration, but there can be no doubt that this aspect—propeller excited vibration—is one which is causing a great deal of concern to-day.

The Author has given a good general survey of the causes of propeller excited vibration and instances propeller clearances as one of the important factors. In the past, propeller clearances have frequently been considered by the ship or propeller designer solely from the propulsion aspect and it is only during recent years that clearances have also been considered as regards vibration. Thus, several of the various criteria proposed, while no doubt suitable for obtaining the desired propulsive characteristics, may not be suitable for avoiding propeller excited vibration. Bunyan's criteria are probably the best available so far as vibration is concerned, but these may well require reconsideration when further experience has been gained with the very large powers now being transmitted on a single screw. The Author correctly says that the provision of adequate clearances does not necessarily imply freedom from vibration, but the converse is almost certainly true. There is little hope of avoiding vibration if clearances are grossly inadequate.

The fact that model tanks are becoming increasingly interested in this problem is to be welcomed since, in the past, the tendency has been to design the after-end lines solely from propulsive considerations. All ship design is a compromise and if the conflicting requirements for vibration and propulsion can be considered at such an early stage as the model tests, then a satisfactory solution may well be reached.

Mr. S. ARCHER

The Association should indeed be grateful to Mr. Lamplough for his contribution to the Transactions on a subject of increasing topical interest. He has judiciously distilled out from the mass of published literature a potent concentrate of useful information which will, I feel sure, be fruitfully absorbed by his colleagues.

There are just one or two remarks I would like to make, some of which may perhaps be regarded as amplifying certain parts of the Author's paper.

First of all, members may be interested to see a photograph of the stern arrangement of the 48,000 ton tanker *Hadrian* (built by A.G. "Weser") mentioned by the Author on page 3. As will be noted, it is not such an extreme form as the Hogner, but nevertheless the measured thrust variation was as low as $\pm 4\frac{1}{2}$ per cent of the mean thrust with even less torque variation. This must be regarded as very creditable when compared with figures such as given in Fig. 2 or Table 1. The latter (due to Brehme) are incidentally generally regarded as somewhat high and were in fact derived from vortex theory without experimental verification. Incidentally, in the case of the s.t. *Hadrian* described by Nitzki (Author's reference (6)) the measured variations of thrust and torque on the full-sized ship were appreciably less than predicted from model experiments, namely, only about 80 per cent and 60 per cent, respectively. It would thus seem that model experiments could only be expected to reveal trends and comparisons with different propellers and stern arrangements rather than to predict quantitative values at full size. Would the Author agree on these points?

Some years ago we received a visit from a well-known Dutch naval architect, who proposed a form of stern arrangement somewhat on the lines of Fig. 10, but with a normal sternframe and rudder. The overhang of the sterntube was almost as great, for which reason the arrangement was colloquially known as the "broomstick" propeller! He claimed that the design had enjoyed considerable success in Western European waters, even under ice conditions as in the River Elbe in winter time. As shown in the sketch (Fig. 1) the tube was formed as a conical shell (Mannesmann Tube Co.) butt-welded to a heavier cylindrical sleeve carried in the sternframe boss proper. Both we, and our colleagues in Ship Plans were, perhaps naturally, somewhat perturbed at the possibilities of damage to the tube by bending from impacts, etc., in service, but after considering the arguments of the designer and taking note of his experience, it was decided to approve the arrangement subject to some strengthening of the welding and attachments to the sternframe bossing. As shown in the sketch (Fig. 2), if P is the impactive force at the tip of one propeller blade, due to, say, striking a

submerged object, then P $\times$ R is the impactive torque on the screwshaft and P $\times$ L is the impactive bending moment on the sterntube at the top of the cone. The designer was able to show that, so far as the tube was concerned, the design was of the "fail safe" type, in that the torsional stress in the shaft due to the torque, PR, would be far greater than the bending stress in the tube under the bending moment, PL, and further, the strength of the propeller blade itself was lower than that of the tube under such an impact.

Before coming to the meeting, I asked C.E.S.R. to look through the records of three LR-classed ships which had been fitted with similar stern arrangements (two with Kort nozzles) and they found no trace of any sterntube or bearing troubles. The ships concerned have the following particulars: —

1. *Garnock,* M/Tug—Kort nozzle— 5 years' service
2. *Brigadier,* M/Tug—Kort nozzle— 6 months' service
3. m.v. *Ponta Garça,* dry cargo—2,950 g.r.t. (as closed shelter decker), Ice Class 3— 10 months' service.

It does seem that this type of stern construction offers good possibilities, particularly in smaller ships and where freedom from vibration is specially important and one would imagine propulsive efficiency should not be inferior to that achievable with the conventional arrangement. That is because, although a lower mean wake would be expected, its adverse effect on efficiency would tend to be counterbalanced by a reduced thrust deduction giving a substantially similar hull efficiency. Further, the propeller efficiency itself would tend to approach closer to the open water value than with a conventional stern form and normal clearances. The Author's comments on this would be of interest.

Turning now to consideration of the Author's remarks on hull vibration and crew comfort, it has fallen to my lot over the years to acquire some experience of such vibrations and their measurement. In particular, I recall three very bad "shakers", all very different types of ships and machinery, viz.: —

1. **H.M.S. "Blyth" and H.M.S. "Peterhead"** (H.M. Fleet Minesweepers)

These were fitted with twin, high speed steam reciprocators of the Yarrow-Schlick-Tweedy, 4-crank, triple expansion type running at about 360 r.p.m. Unfortunately, the weights of reciprocating parts did not entirely agree with the design figures and in consequence there was a residual primary unbalance at operating speed (which had had to be reduced to 300 r.p.m.) amounting to 0·84 tons force and 11·83 tons/ft. couple per engine. Additionally, there was a substantial secondary couple of about 56 tons/ft. per engine.

The vibration on board at or near the various resonances was truly formidable, in fact, the ships were almost non-operational in consequence. Old "salts" on board said they had never sailed under conditions anywhere near so bad and complained of all sorts of ailments, even including sea sickness, and one actually claimed he had difficulty at times in retaining his dental plate!! These ships were eventually made habitable by bolting on suitable additional small balance weights on the crank-webs to reduce the primary vibration and by the fitting of harmonic balancers at the forward and after ends of each engine, chain-driven at twice engine speed from the main shafting just abaft the engines to reduce the secondary vibration.

2. **S/S "Belevelyn" and S/S "Viktun"**

The second case related to two single screw cargo sister ships which had several novel features, including machinery right aft comprising 4-crank, double compound, Woolf type steam reciprocators. Unfortunately, the engine was inherently unbalanced and there was a large primary couple which excited resonant 2-node vertical vibration at about 90 per cent of the full service speed of 115 r.p.m., when in the trial condition at about half full, load displacement. The resulting vibration on the *Viktun* reached such a high amplitude that the starboard bulwark bar actually fractured with a report like a gun, much to everyone's concern! In this case the vibration was cured by redesigning the built-up crankshaft with large integral balance weights on the webs.

3. **H.N.M.S. "No. 231"**

This was a single-screw wooden minesweeper of the Royal Dutch Navy fitted with an 8-cylinder 4-S.C. diesel engine and a 3-bladed propeller running at a normal speed of 325 r.p.m. The vibration in this instance was almost entirely propeller-excited at third order in the athwartships horizontal plane, and at sixth order in the vertical plane, in both cases approaching resonance on the lower flanks near the service r.p.m. The vibratory conditions in the crew's quarters right aft were most unpleasant, the third order horizontal vibration being particularly distressing. The remedy here was fairly obvious, namely, a change from a 3-bladed to a 4-bladed propeller, to bring the resonant critical speed well below the normal service r.p.m.

Details of the measured vibration frequencies, amplitudes, vibratory velocities and accelerations together with other data, including (under "Remarks") a subjective assessment of the degree of severity of the vibratory conditions, are given in the Table (Fig. 7).

The various maximum vibration velocities and accelerations are plotted log/log against frequency in cycles per second on the accompanying graphs (Figs. 3 and 4). On the same graphs are drawn a number of lines due to Reiher & Meister* based on experiments on human subjects exposed to vertical vibration and which purport to represent conditions of constant degrees of tolerance or unpleasantness. On the graph (Fig. 3) are also shown a number of lines of constant velocity in

the range of 50 to 100 cycles per second corresponding to Yates's† suggested criteria of degrees of rough running in steam turbine rotors, as subjectively assessed from bearing pedestal vibration. On the frequency/acceleration graph (Fig. 4) are drawn several other proposed constant vertical acceleration criteria, including that due to Hinsonϕ for limiting crew comfort, by Mallock‡ at 5 per cent g for "Unpleasant" vibration and by the Author for "Unbearable" vibration (after Dieudonné)¶. For comparison with the data of H.N.M.S. *No. 231* in the above Fig. 4, a vertical line has been included representing the Author's suggested "Unbearable" limit for horizontal vibration (also after Dieudonné).

Certain general conclusions emerge from these diagrams namely:—

I. If Reiher & Meister's curves of constant vibration "intensity" are accepted, then

(i) Greater vibration velocities can be tolerated at low frequencies than at higher frequencies, and

(ii) Higher vibration accelerations can be tolerated as the frequency rises, and

II. Most of the plotted points for the ships given in the Table lie to the right of the "Dangerous" lines, whether the criterion is vibration velocity or acceleration, and the remainder at least lie to the right of the "Disagreeable" lines. The conclusion is therefore clear that the ships cited were subjected to unusually severe vibratory conditions by all known standards, quite apart from my own subjective assessment of their severity.

The problem of crew, and especially passenger comfort is indeed difficult to define. For one thing, the human body does not exhibit the same degree of tolerance to acceleration or velocity at all frequencies and in this respect resembles the behaviour of the human ear whose assessment of equal loudness varies in a somewhat complex manner with frequency and is by no means constant. For another, it is perhaps not generally appreciated that, under vertical "whole body" vibration, human beings exhibit resonance within the range of 200-300 c.p.m. ($3\frac{1}{2}$ to 5 c.p.s.)§. This, of course, comes well within the range of propeller blade excitation frequencies and may partly account for the reduced human tolerance of vertical vibration over this frequency band and increasing tolerance above it. In this respect, it may well be significant that the particularly unpleasant vertical vibration due to primary engine unbalance on both H.M.S. *Peterhead* and H.M.S. *Blyth* (*see* Table and Fig. 4) had a frequency between 4 and 5 cycles per second.

* Reiher & Meister—"Akustische Zeitschrift", 1937.

† H. G. Yates, M.A.—"Vibration Diagnosis in Marine Geared Turbines". Trans. N.E. Coast Inst. of Engrs & Shipbuilders Vol. 65, 1949.

ϕ A. R. Hinson—"Some Notes on Vibration Problems". Staff Association Paper, 1960/61 (2).

¶ J. Dieudonné—Author's reference No. 12 "Vibration in Ships". TRINA 1959.

‡ F. Postlethwaite—"Human Susceptibility to Vibration"—"Engineering" Vol. 157 (1) 1944.

§ "Some Physiological Effects of Low-Frequency, High Amplitude Vibration". M. A. Schmitz and C. A. Boettcher.—A.S.M.E. Pnt. No. 60-PROD-17.

As regards horizontal vibration it is worth noting that on the graph (Fig. 4) the acceleration due to the third order propeller-excited vibration on H.N.M.S. *No. 231* slightly exceeds the Author's constant value for "Unbearable" vibration (after Dieudonné). It was certainly highly unpleasant for all concerned, but perhaps hardly unbearable.

From the aspect of hull structural damage due to vibration effects, the problem of defining acceptable limits bristles with difficulties. Hinsonϕ quotes various figures which have been given in the literature, e.g. 10 per cent g (3 ft./sec.^{2}) for the maximum vertical vibration acceleration, presumably at the after anti-node. In my opinion, such arbitrary limits cannot hope to give any reliable guidance without, in some way, linking frequency, amplitude, velocity or acceleration with resulting additional fatigue stress, with some attempt also to allow for stress concentration at discontinuities, notches, etc. The fact that on the s.s. *Viktun* the acceleration at the after anti-node, when the bulwark bar fractured, was about 6 ft./sec.^{2} and at the point of damage some 2 ft./sec.^{2} was doubtless fortuitous, although of the same order of magnitude as the 10 per cent g limit. The Author's further thoughts on this difficult subject would be welcome.

Vibration excited by propeller thrust variation could have severe effects on main thrust bearings, thrust seatings and reduction gears, etc., should resonance occur near the service r.p.m. of the propellers and in this respect, the 4-blader was usually the worst culprit, especially where aperture clearances were tight, after-body lines too abrupt or propeller blade shapes unsuitable. The results of recent tests carried out by Johnson and McClimont of B.S.R.A. are given in Fig. 5, from which it is seen that although the amplitude measured at the thrust block was only $\pm$ 0·010 in. ($\pm$ 0·25 mm.), this actually corresponded to a thrust variation of some $\pm$ 23 tons, or about $\pm$ 40 per cent of the mean full-power thrust at the service speed of 120 r.p.m., which represents a dynamic magnification of the propeller excitation amounting to about three and a half only. The service speed was at 80 per cent of this 1-node, fourth order resonant speed. The point to note is that although the thrust block amplitude was barely detectible to the naked eye, nevertheless, owing to the stiffness of the block/seating combination, namely, about 2,300 tons per inch, this represented a very heavy load variation on the structure. Thus, to base liability to structural damage on amplitude or acceleration alone may clearly be highly misleading to the unwary. It would be interesting to have the Author's views on what maximum percentage full-power $\pm$ thrust variation at the service speed he would consider appropriate from a seating structural damage point of view.

MR. J. H. MILTON

Mr. Lamplough has written a very interesting and informative paper on a subject, which, judging by troubles being experienced, does not appear to

receive sufficient consideration prior to building ships.

The gist of the opening two sentences of this paper perhaps has some bearing on the prevalence of such troubles—Mr. Lamplough states—ship vibrations of one sort or another have troubled most Naval Architects and that these vibrations of the hull are either engine or propeller excited—thus we have "troubled" Naval Architects looking to Engineers for vibrationless machinery, and propellers which will operate smoothly in troubled waters! One might venture to suggest that more co-operation between Naval Architects and Engineers at the design stage might produce vessels less likely to vibrate at their after ends—especially when, as quoted on page 2, Wake Variation is the principal cause of the type of vibration under discussion.

Many years ago whilst a Junior Surveyor I was the owner of a Riley 9 car, the 4-cylinder engine of which, despite all my efforts, refused to run quietly. The reason was that the two camshafts, each with four cams, were gear driven from the crankshaft and the torque absorption properties of these camshafts caused the driving gears to hammer heavily.

As a poor analogy maybe, the dead smooth torque from our modern high efficiency turbine installations (with their gears hobbed and shaved to the smallest of limits, wheels and rotors very finely balanced) is fed into a four- or five-bladed propeller operating in, or partly in, very troubled water.

When investigating the type of vibration under consideration in this paper and the patterned wear of tailshaft liners sometimes associated with it, recommendations have to be made for eliminating the cause and it is at this point where difficulties arise as, firstly, variations in wake speeds are a function of the original hull design and apart from adding water-deflecting appendages to the hull (which even the boldest would shrink from recommending), there is little that can be done once the vessel is in *being*. Secondly, redesigning the propeller, perhaps with the loss of some efficiency, so that with a different number of blades, greater skewback, different shaped blading and trailing edges, etc., is a possible but expensive remedy. It would operate more smoothly under varying wake conditions, thus avoiding abrupt changes in thrust balance—always bearing in mind that changing from four to five blades increases the frequency of the exciting force accordingly and thus in its turn could bring a resonant condition down into the speed range.

Mr. Lamplough mentioned on page two methods of assessing the forces producing the amplitudes measured on vessels—and it would be interesting to know whether vibration generators so phased to counter propeller excited vibration have ever been used on board ship.

Finally, as wake variation is the principal cause of the type of vibration discussed in this paper, perhaps Mr. Lamplough could enlighten me as to the effect, on a ship with propeller excited after-end vibration, when going astern with the rudder amidships.

Mr. W. BLACKLOCK

The Author does not discuss the prediction of hull natural frequencies and perhaps he feels that this subject has been adequately dealt with elsewhere. If this is so and he has no criticism to offer against the methods of determination of natural frequency then perhaps there is a case for the Society to establish Rules or Guidance Notes regarding hull criticals in much the same way that the Machinery Rules cover torsional criticals.

This may help to avoid major hull vibration but it would seem almost impossible to cover local resonance which is an indeterminate factor in ship design. It appears therefore that we either have to live with local resonance or stiffen locally depending upon the acceleration figure of 3 ft./sec.^{2} which is the only criterion of structural damage. I think the Author will agree that this is a purely arbitrary value and it would be of interest to have any information relating structural damage to vibration amplitudes and frequencies.

Something in the nature of Table No. 2 in the paper would be desirable since the Society officially has no feeling for the comfort of the crew but is primarily concerned with structural damage. Indeed the biggest noise producer in a turbine-driven ship could be the reduction gearing which can make life unbearable in the engine room. However, it is not always the case that a noisy gear is a source of mechanical trouble and some of the quietest gears have had disastrous failures.

Finally, it is of historical interest to note that vibration troubles originating from the shape of the after body which have come to light in recent years may have some connection with the adoption of cruiser sterns. Perhaps a reversion to the old-fashioned counter stern may solve our problems.

Mr. N. FLENSBURG

Mr. Lamplough is to be congratulated for his most interesting paper regarding propeller excited vibration. I propose to confine my remarks to the effect on the after body of the ship, when a propeller is used in conjunction with a Mariner type under-hung rudder.

A Mariner type rudder is fitted partly because, with the aperture arrangement associated with this type of rudder, the wake variation is smaller and therefore the propeller vibration is reduced. It seems, therefore, unfortunate that there recently has been rather serious vibration trouble in the after body of large tankers fitted with Mariner type rudders. This trouble occurs in ships with motor or turbine drive, and could not be referred to under the heading "general hull vibration". The cause seems to be the propeller itself. The result is fractures in the A.P. tank, counter and also the steel structure forward and above the after peak.

It seems to be extremely important that the structure of the rudder horn supporting the Mariner type rudder is well built into the after peak structure by means of heavy vertical floors and side webs and that these vertical members are tied together by horizontal flats and side stringers. Together with the hull these vertical and horizontal members will take up the static loads from the rudder horn and also reduce the vibratory loading.

In my opinion, no scallops should be allowed in floor, web and frame connection to the hull in the after peak and counter. This should be the case not only in ships with Mariner type rudders but also in certain other ships where power and speed are higher than normal. Further, in these types of ships corrugated bulkheads should be avoided in way of, and in the vicinity of the A.P. tank. There have been various fractures in this region, which, as far as is understood, were partly caused by vibration.

In order to reduce the vertical vibration above and forward of the A.P. tank, it has been decided, on one of the large tankers fitted with Mariner type rudder, to instal two longitudinal bulkheads, which extend from frame 0 to 20 in the longitudinal direction and from the shell to the main deck in the vertical direction. By fitting these extra bulkheads it is hoped that the rather extensive vibration found in this region will be cured in the sister vessel.

AUTHOR'S REPLY

To Mr. J. B. Davies

The increased interest shown in vibration problems has been stimulated in part by the results of fitting larger engine powers in association with single screws. An example of the effect of this is given in the case of the Great Lakes vessel quoted in the paper. Modern single screw vessels fitted with engines upwards of 12,000 h.p. are mainly confined to the tanker, bulk carrier types, and it is not surprising therefore that these types seem to suffer more from propeller excitation and hence, open type stern arrangements are more usually seen on these ships. It could well be as, Mr. Davies suggests, the present-day acceptable clearances for normal sternframes are not adequate for these particular types and may therefore have to be reassessed. It is to be hoped that the testing tanks can produce more definite information on this subject.

To Mr. S. Archer

I am indebted to Mr. Archer for the photograph of m.t. *Hadrian* (Fig. 6), which clearly shows the pronounced bulbous effect and the hollowing out of the lines forward and above the propeller.

It is agreed that the use of models will not predict quantitative values at full size but, despite this, they indicate the general effect of a change in shape aft on the wake distribution, as seen for example in Figs. 4 and 5 of the paper. They are therefore a valuable aid to determining the optimum hull form aft from the vibration viewpoint.

Mr. Archer's comments on the strength of the "broomstick" sterntube arrangement are enlightening. At first glance it appeared rather horrible when seen through a plans surveyor's eyes.

The argument that the open type stern arrangement should not be inferior to the normal type from the propulsive efficiency viewpoint is substantiated generally by the results given in Ref. 5 of the paper. For the Hogner model a propulsive coefficient of ·77 was found when running at a ship speed of 16 knots, while for the moderately U-shaped model, with normal stern arrangement, a value of ·74 was obtained. These values are not strictly comparative, as the propeller used on the Hogner model was somewhat different in design from that used on the other model, but they indicate that, provided the propeller is appropriately designed, no large difference in propulsive efficiency should occur due to the open type stern arrangement. One other thing which is of interest here, is that the power absorption characteristics of the Hogner form resulted in 3 per cent more power being required to achieve the same service speed as compared with the moderately U-shaped hull, i.e. the designer is obliged to pay slightly for his better vibration qualities.

On the subject of what levels of vibration may be considered acceptable and which criteria should be used to assess these levels, it is clear that the use of an acceleration value is arbitrary. The values which are considered acceptable vary between authorities, but the least that may be said for acceleration as the criteria is that it is a simple one and easily obtained.

It is suggested that some method of linking frequency amplitude, velocity and acceleration with resulting fatigue stress would give a more reliable guide. On attempting to do this it will be found that, allotting a realistic value to the fatigue stress acceptable for general use on a complex structure like a ship is fraught with difficulty; pre-stressing effects due to welding, shape and hence inertia of member and notch effects, are only a few items which affect this choice. Thus the chosen stress itself will then be of an arbitrary nature and as such will be as open to attack as the acceleration criteria.

Having had no experience of troubles in thrust seatings due to thrust fluctuations I cannot offer any values for permissible fluctuations of thrust. Enquiries in the Gothenburg Office tend to show that little trouble is found in these seats (this may well be borne out by the fact that the Society only requires that they be inspected at Special Survey). One case was reported, however, and this was on a single screw tanker fitted with a turbine giving 20,000 s.h.p. The cause of the trouble was not stated, but the thrust block was seen to be tipping in a fore-and-aft direction. The block was removed and the seat considerably re-designed and reinforced, this resulted in a substantial improvement.

To Mr. J. H. Milton

Naval Architects have accepted, with reasonably good grace, the fact that propelling machinery is a necessary evil. However, Mr. Milton's suggestion for more co-operation at the design stage is whole-heartedly agreed to. As he rightly points out, little can be done once the vessel is in being. Anything which can be done will be expensive and not 100 per cent certain to achieve a complete cure, therefore it is imperative to take all possible precautions against this form of trouble at the design stage.

An "anti-vibration" device used on a vessel is described by Constantini*. The apparatus consisted basically of a mass which was a cast iron tank filled with water, attached to a rigid frame by heavy springs. The frame was built into the ship's structure at an anti-node (in the case described this was at the aft end of the vessel). This system was excited by any ship hull vibrations and the mass could be phased 180° to the exciting force, thus providing an opposing vibratory force of the same frequency as the disturbing force. With the system in action it was claimed that a reduction in amplitude of 94 per cent was obtained.

When a vessel runs astern with the rudder amidships, the wake distribution at the propeller disc will obviously be different from the ahead condition and should tend to be homogeneous for the purpose of this discussion. If a vessel suffers from propeller excitation in this condition then it is probably a result of too small a clearance between the propeller and the top of the aperture. Another factor to be remembered is that, for some vessels, e.g. fairly large tankers, the first natural hull critical frequency lies below the normal running speed, and therefore when running astern at say 60–80 per cent normal shaft r.p.m., any small excitation from the propeller may be at the same frequency as the hull and the effect magnified by resonance.

To Mr. W. Blacklock

My remarks in the paper are confined mainly to the local aspect of propeller excited vibration. Mr. Blacklock's suggestion that there is a case for the Society to give guidance in the matter of hull criticals is reasonable, provided that the means are available to establish, with sufficient accuracy, the main critical frequencies of the hull. A hull critical is usually sharply tuned and the accuracy of the method used to predict the critical frequency should therefore be of the order of ± 1 per cent.

One of the more accurate methods, due to Prohaska, predicts frequencies with errors of the order of ± 2–3 per cent, but this is for the first mode of vertical vibration only. It may be that with the increasing use of computers for this type of work, sufficient accuracy will be forthcoming, then possibly the Society could formulate some form of guidance notes for builders, on this problem.

It is agreed that the value for acceptable acceleration is purely arbitrary and should be used only for guidance.

Fig. 11 in the paper shows a vessel which had trouble and had a counter stern. The significant thing here is that she was in service for several years with no trouble, the engine power was increased some 50 per cent then serious trouble began.

To Mr. N. Flensburg

Mr. Flensburg's comments on the necessity of integrating the underhung rudder structure into the main hull are agreed to. Whether, in the case mentioned, the trouble was attributable directly to the underslung rudder or to a general weakness in the adjacent structure was not clearly apparent, however, many fractures in the after peak region, particularly in the horizontal stringers fitted, indicated that considerable rearrangement of the design was necessary.

For these large overhanging sterns, one shipyard here considered it necessary, as Mr. Flensburg says, to fit two longitudinal bulkheads, in line with the casing and extending as far aft as practicable, to form a deep longitudinal girder to support the overhang. The vibration results from the first ship fitted with these bulkheads compare favourably with a sister ship which did not have them.

In conclusion, I would like to thank those people who contributed to the discussion.

* Vibration in Ships", M. Constantini, R.I.N.A. 1938.

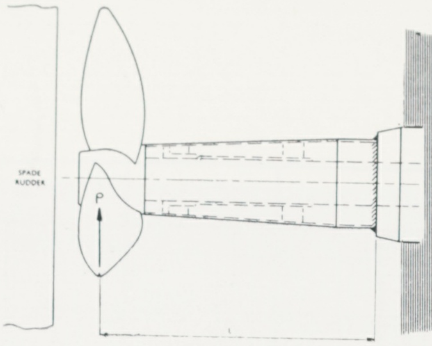


FIG. 1

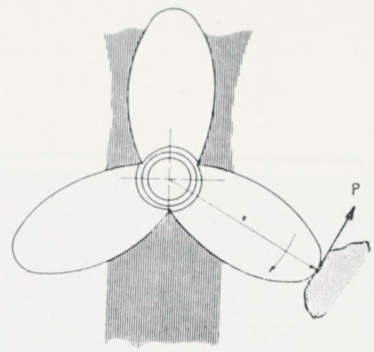


FIG. 2

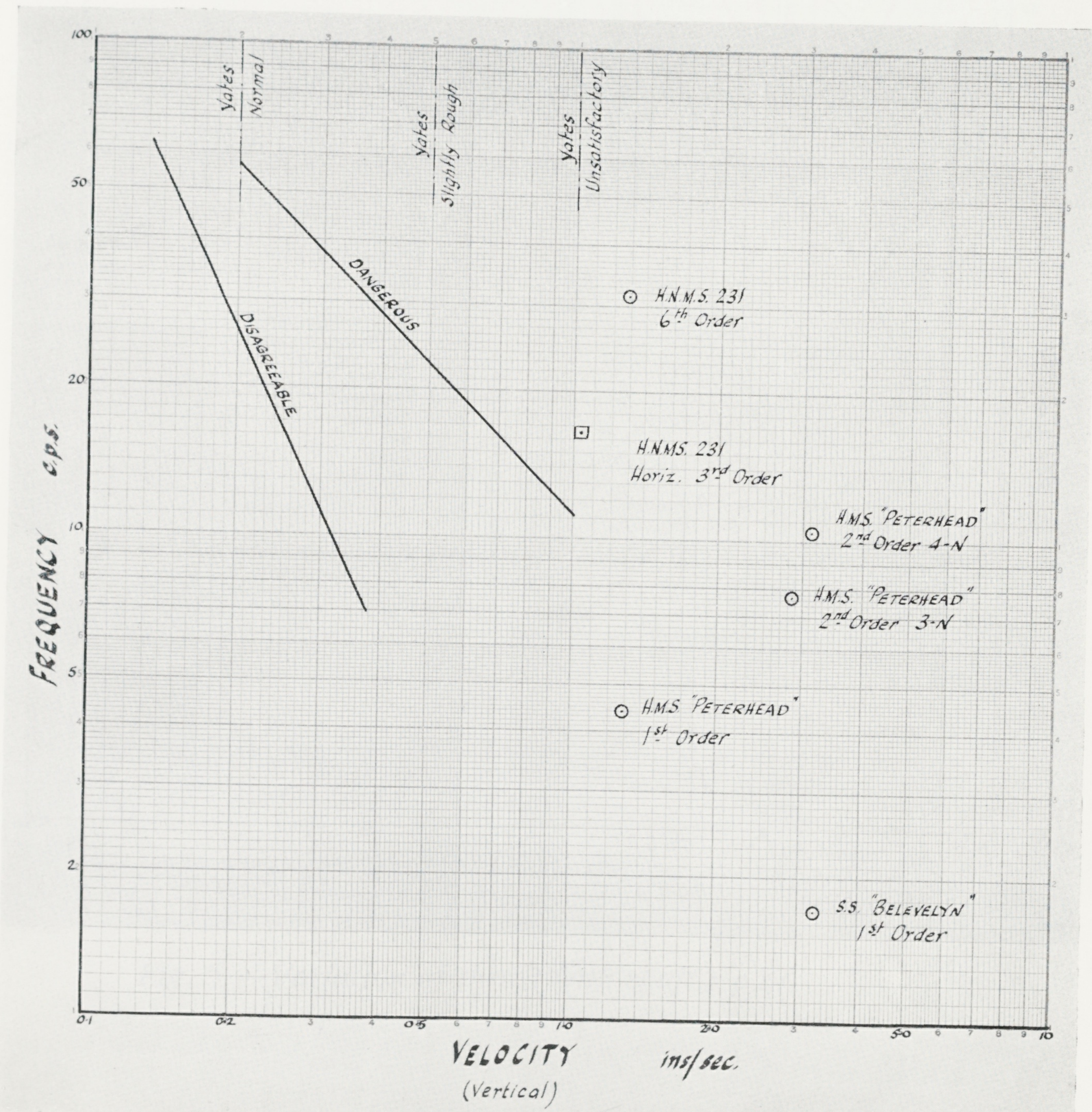
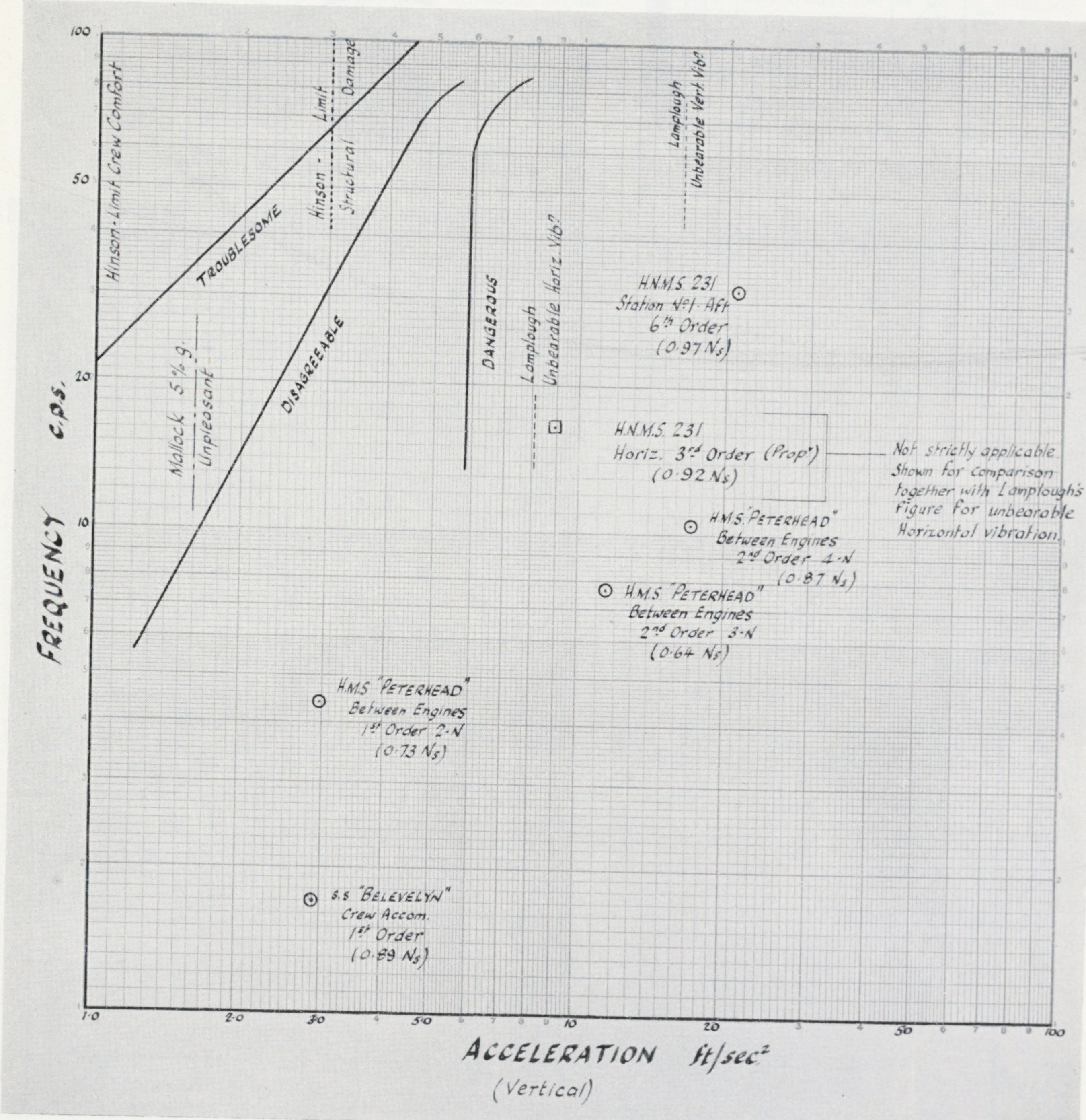


FIG. 3



Hinson - Limit Crew Comfort
TROUBLESOME
DISAGREEABLE
DANGEROUS
Hinson - Limit
Structural Damage
Mallock 5% g.
Unpleasant
Lamplough
Unbearable Horiz. Vib?
Lamplough
Unbearable Vert. Vib?
H.N.M.S. 231
Station No 1 Aft
6th Order
(0·97 Ns)
H.N.M.S. 231
Horiz. 3rd Order (Propr)
(0·92 Ns)
Not strictly applicable
Shown for comparison
together with Lamplough's
figure for unbearable
Horizontal vibration
H.M.S. "PETERHEAD"
Between Engines
2nd Order 4-N
(0·87 Ns)
H.M.S. "PETERHEAD"
Between Engines
2nd Order 3-N
(0·64 Ns)
H.M.S. "PETERHEAD"
Between Engines
1st Order 2-N
(0·73 Ns)
S.S. "BELEVELYN"
Crew Accom.
1st Order
(0·89 Ns)
FREQUENCY c.p.s.
ACCELERATION ft/sec²
(Vertical)

FIG. 4

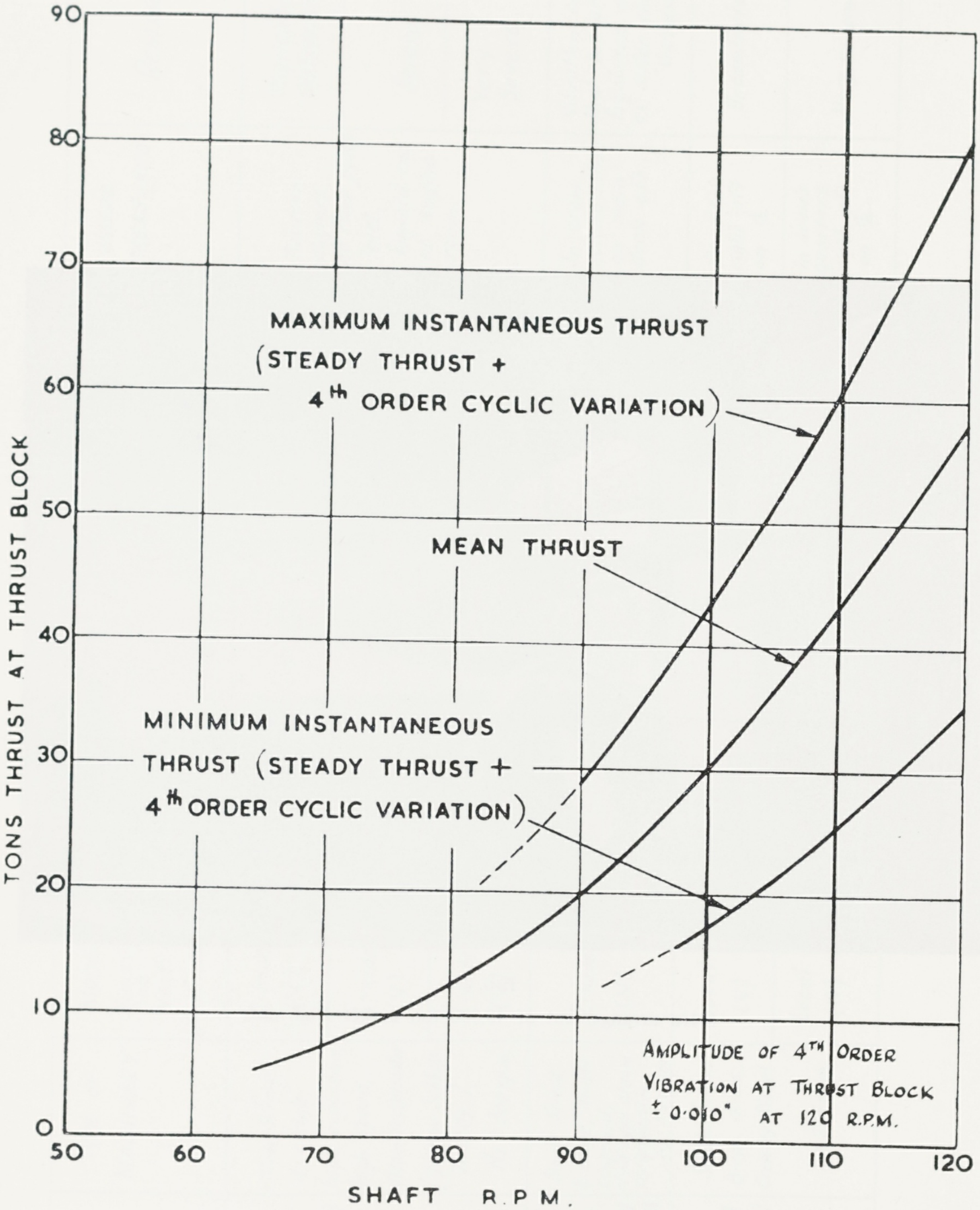


Fig. 5.—Variation of thrust with shaft r.p.m.

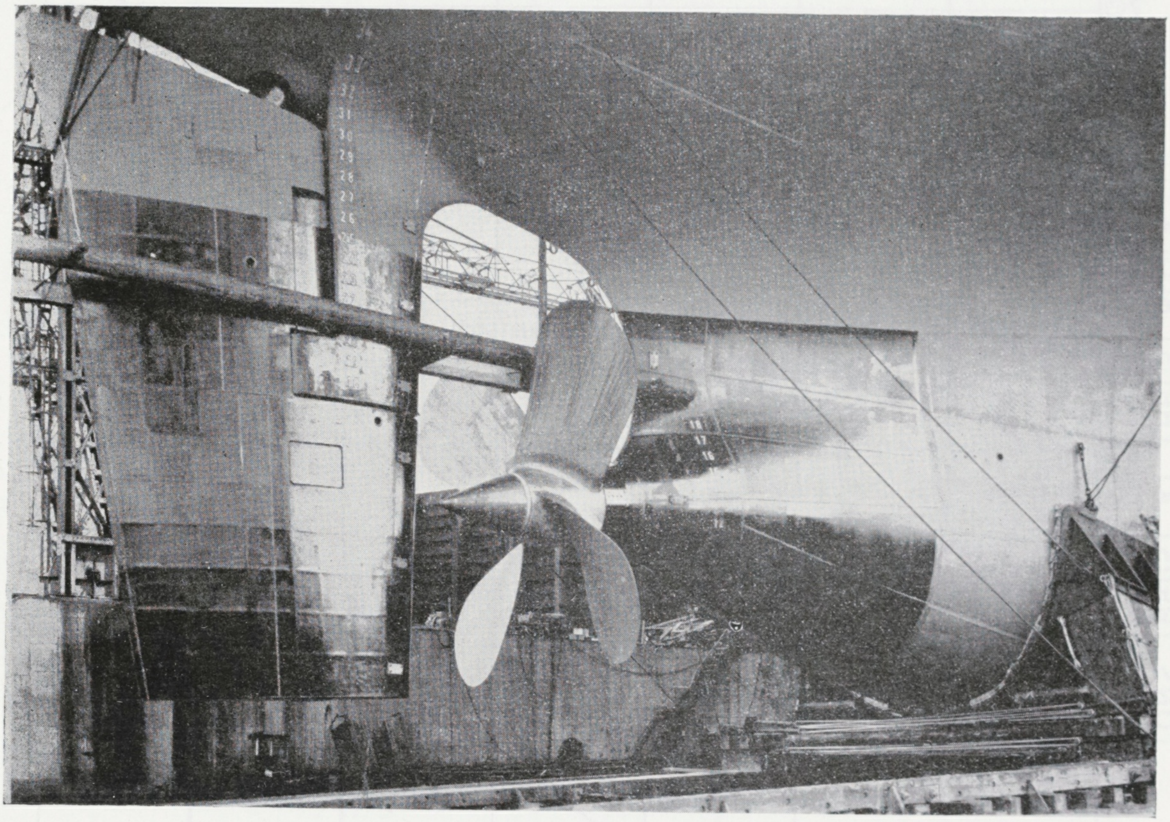


FIG. 6
Stern arrangement of 48,000 ton tanker *Hadrian*

NAME AND TYPE OF SHIP	TYPE OF MACHINERY	MODE Horiz. or Vert. (No. of modes)	ENGINE R.P.M.	ORDER i.e. Cycles/rev.	FREQUENCY (c.p.s.)	VIBRATION AMPLITUDE (± inches)	MAXIMUM VIBRATION VELOCITY (± ins/sec)	MAXIMUM VIBRATION ACCELERATION (± ft/sec²)	WHERE MEASURED	REMARKS.
H.M.S's. "BLYTH" & "PETERHEAD" (Fleet Minesweepers) Riveted Hulls including engine seatings. Engines Aft.	Twin Screw, 4-crank, triple expansion, high speed, steam reciprocator (Yarrow-Schlick-Tweedy) N_s = 300 r.p.m.	2-node (V)	261·6	1st.	4·4	0·047	1·3	3·0	Between engines, at floor plate level, forward end of engine room.	Physically Distressing.
		3-node (V)	230	2nd.	7·7	0·059	2·85	11·5		Severe.
		4-node (V)	314	2nd.	10·5	0·047	3·1	17·1		Very Severe.
S.S. "BELEVELYN" 4780 g.r.t. Dry Cargo, Eng. Aft. Displ. 5100 tons (trials) 10920 - (loaded)	Single Screw, 4-crank, double compound, Woolf type steam reciprocator N_s = 115 r.p.m.	2-node (V)	102·5 (0·9 N_s)	1st.	1·7	0·3	3·25	2·9	In crew accomodation on main deck aft.	Vibration so severe that bulwark bar fractured on trials of sister ship. "VIKTUN"
H.N.M.S. No 231 Royal Dutch Naval Ship (Minesweeper) Wooden Hulled. 285 tons displ.	Single Screw, 8-cyl. 4-S.C. Crossley Diesel N_s = 325 r.p.m.	Local (V)	315	6th.	31·5	0·0065	1·3	21·2	On deck right aft on ℄.	Uncomfortable.
		Local (H)	330	3rd.	16·5	0·010	1·04	8·94	In crew's quarters right aft on ℄.	Very uncomfortable.

FIG. 7

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at Garrett House
Manor Royal, Crawley, Sussex, England

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Staff Assocation

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THE MACHINERY OF THE "GREAT EASTERN"

by

J. GUTHRIE

LLOYD'S REGISTER OF SHIPPING
71, Fenchurch Street, LONDON, E.C.3

The Machinery of the "Great Eastern"

By J. Guthrie

Apologia

Never in all history has any ship captured the public imagination as did the *Great Eastern* in the 1860's: the eighth wonder of the world, the fabulous record breaker of the North Atlantic, the toast of two continents, the monster that ran through fortune after fortune and destroyed builders and owners with grim impartiality, the ship that spent most of her life at anchor.

There must be few Englishmen living in the second half of the nineteenth century who had never visited, seen or heard of this great ship, and in that unsophisticated age, the man-in-the-street probably knew a great deal more about the *Great Eastern* than his present-day counterpart knows about the *Queen Mary*. Yet, in spite of this, she rusted away without leaving any record for posterity, and the lack of technical knowledge about her is almost uncanny.

However, odd clues have been followed to museums and dingy little book shops, and sufficient information has been collected to produce a fair description of her machinery from the writings of some of the people who came in contact with her. These include the builder, the draughtsman, an engineer consultant who advised in the design, a naval engineer, the author of an engineering manual for B.O.T. examinations, and sundry eminent engineers of the period who were consulted by the owners.

These Victorian engineers had a neat turn of phrase and were masters of the arts of draughtsmanship and engraving. Furthermore, they wrote from personal experience, and it is therefore proposed to let them describe the various items of machinery in their own words as far as possible.

Brief History

The *Great Eastern* was launched in January, 1858, carried out trials in September, 1859, and sailed on her maiden voyage to New York in May, 1860. On the return from this voyage, her screwshaft gave out, and a new shaft was fitted the following winter lay-up. On her third voyage she carried troops to Quebec and on the fourth voyage, in early 1862, she lost both paddle wheels and broke her rudder head in a storm in mid-Atlantic. In August, 1862, the vessel struck the Great Eastern Rock approaching New York and ripped the outer shell in way of the starboard bilge for a length of 83 ft. This hole was repaired at New York by means of a cofferdam, there being no dry dock in the world suitable for a vessel of this size. She was laid up in 1864 through lack of funds to run her, but was engaged sporadically between 1865 and 1874 to lay cables across the Atlantic and to India. The French Government chartered her for one voyage to New York in 1867, but from 1874 onwards she was bought and sold some half-dozen times without ever sailing. In the Spring of 1886 she was purchased by a large department store in Liverpool and was anchored in the Mersey as a showboat-cum-billboard. After a brief visit to Dublin and Greenock, she was broken up in Birkenhead in January, 1889.

Particulars of the Vessel

Length B.P.	680 feet
Breadth over sponsons	120 „
Depth	58 „
Gross tonnage	18,914 tons
Displacement	27,384 „
Speed	$14\frac{1}{2}$ knots
Keel laid	1-5-1854

The *Great Eastern* was the "mostest" vessel ever built, being: The only vessel to have five funnels; the only vessel to have screw, paddle and sail propulsion; the only vessel to have two entirely independent sets of propelling machinery each with its own set of boilers, its own chief engineer and engine-room personnel housed in separate parts of the ship; the first merchant steamer to have her master knighted; the biggest vessel ever built at that time. (The largest vessel in existence at the time of her launching was the *Himalaya*, 340 ft. long, i.e., exactly half her length. The first vessel to exceed the *Great Eastern* in tonnage was the *Lusitania*, built in 1906); the ugliest vessel afloat.

To make up for this last point, she was also the safest of ships, as she never lost a passenger during the whole of her career. The 1860's was an age of recklessness when ships and machinery were driven across the Atlantic for all they were worth: "pillars of fire by night and pillars of smoke by day", and comparatively few of them lived to be broken up. During this period, the *Austria* caught fire in mid-Atlantic and sank with 492 people on board; the *Atlantic* went ashore off Halifax and was lost with 560 on board; the *Arctic* was in collision in fog on the Banks and sank with 322 souls, while the *Pacific* disappeared in the N. Atlantic with 159 passengers and crew. In spite of her good safety record, however, the *Great Eastern* never paid her way, and after many years of laying up she ended her days as an advertisement for a Liverpool department store.

This vessel is always spoken of as having been designed and built by I. K. Brunel, but the facts hardly bear out this idea. She was designed by J. Scott Russell, erstwhile professor of natural philosophy at St. Andrews University, on his wave-line principle, and built by him at his shipyard at

Millwall on the Thames. The screw engines and boilers were supplied by J. Watt & Co., Birmingham, and the paddle engines and boilers were built by J. Scott Russell & Co. from the patent taken out by Joseph Maudslay in 1827.

Several ideas were undoubtedly Brunel's, such as providing bunker capacity for 12,000 tons of coal to enable the vessel to make a round voyage to Australia without bunkering, the cellular system of double bottom tanks and longitudinal framing, providing two separate means of steam propulsion, and finally superintending the launching arrangements. Probably Brunel's greatest asset was in promoting the company to finance the building and supplying the nervous energy to see the construction to a satisfactory conclusion.

Launching

The launch was carried out sideways and was a total failure, the vessel having to be pushed every inch of the way into the water. The launching date was fixed for 3rd November, 1857, but the *Great Eastern* was not afloat until 31st January, 1858. C. H. Jordan, a draughtsman at the shipyard, who took an active part at the launch, gives us this description of the technical details: "The vessel was built broadside to the river. Two launching ways were constructed for launching her, each about 80 ft. in width and about 140 ft. apart, laid at an inclination of about 1 in 12. They had railway lines screwed on them, 8 in. apart, with their surfaces coated with black lead. The under side of the launching cradles were shod with rolled iron bars 7 in. wide by 1 in. thick, 12 in. apart, and fixed at right angles to the bars on the launching ways. The use of two iron surfaces in contact, and the consequent friction between them, due to the weight of the vessel, did not appear to have been sufficiently reckoned with by Mr. Brunel, nor did he appear to have foreseen the impossibility of the vessel proceeding down each of the two slipways, in launching, at identically the same speed; and that the inequality that was sure to occur would cause the vessel to proceed down the ways, on the cradles, in an oblique direction and possibly produce failure in launching"

That is precisely what did occur.

By contrast, successful side launchings are fairly common in U.S.A., and on the Great Lakes a vessel the size of the *Great Eastern* would be launched in the following manner:—

The ship having been built close to and parallel to the water's edge, 50 or 60 ground ways are constructed below the keel and at right angles to it. These ways are about 24 in. wide and 10 ft. apart, and upon each one, a sliding way or slipper is fitted which supports the hull through crib work. Groups of these sliding ways are tied together to form an integral structure. The half dozen sliding ways at each end are the trigger ways to hold the vessel before launching, while the next half dozen at each end form the guide ways, i.e., their ground ways are fitted with ribbands to prevent fore and aft movement.

The declivity of the ways varies from $1\frac{3}{16}$ to 2 in. per foot, and the standing and sliding surfaces are covered in the usual manner with a layer of tallow, grease and soap.

When the keel blocks are removed, the vessel will normally drop a fraction of an inch and hang on the triggers, and should therefore start as soon as the triggers fall. To make certain of success, however, some strain is usually taken on the jacks forward and aft before tripping the triggers.

Machinery

It is only in recent years that diesel engineers have attained that elusive figure of 2,000 h.p. per cylinder, at one time considered to be the ultimate in design of reciprocating machinery. Yet a hundred years ago this power was achieved in the paddle steamer *Scotia* developing 4,600 h.p. in two cylinders with only 25 lb. boiler pressure. When required to produce a more powerful engine, our ancestors had no choice but to tackle the problem in the only way known to them: to scale up the machinery to a monstrous size. This trend could be followed in the passenger ships just before the advent of the steam turbine, and some readers may remember the wing engines of the *Olympic,* probably the most powerful marine steam reciprocating engines ever built. These four-cylinder giants had a stroke of 6 ft. 3 in.; the imagination boggles at a four-cylinder engine with a stroke of 14 feet! Fig. 1 shows an elevation of the *Olympic's* engine and the *Great Eastern's* paddle engine drawn to the same scale.

Paddle Engine Particulars

Oscillating engine, four cylinders 74 in. diameter by 14 ft. stroke, inverted V type, developing 3,410 i.h.p. at 10·75 r.p.m. giving eight knots on paddles alone. Weight of engine: 836 tons.

The following description is by J. Scott Russell, see Figs. 2 and 3: "It will be observed that these engines rest on four great beams, which run the whole length of the 40-ft. engine room. These beams rise 14 ft. above the floor, and are, like the rest of the internal work of the engine room, cellular bulkheads of $\frac{1}{2}$ in. plate and angle iron. These beams are about 10 ft. apart, and divide the engine into three portions; viz. a pair of oscillating engines on the left, a pair of oscillating engines on the right, and the air pumps in the centre. It will be observed that each pair of oscillating engines is coupled to a single crank-pin, an arrangement in favour of which I have elsewhere avowed my strong partiality. The working of the engines is brought to the centre, and they are handled from a platform immediately above the air pumps, which are worked by a crank in the intermediate shaft. The two cranks on the end of the intermediate shaft differ in no respect from the ordinary crank, and carry a crank-pin on which the two engines work. It may be noticed that there is no second crank to work the paddle shafts; but instead, there is a large wheel of cast iron keyed on the outer shaft, embraced by a friction strap, and into an eye of that

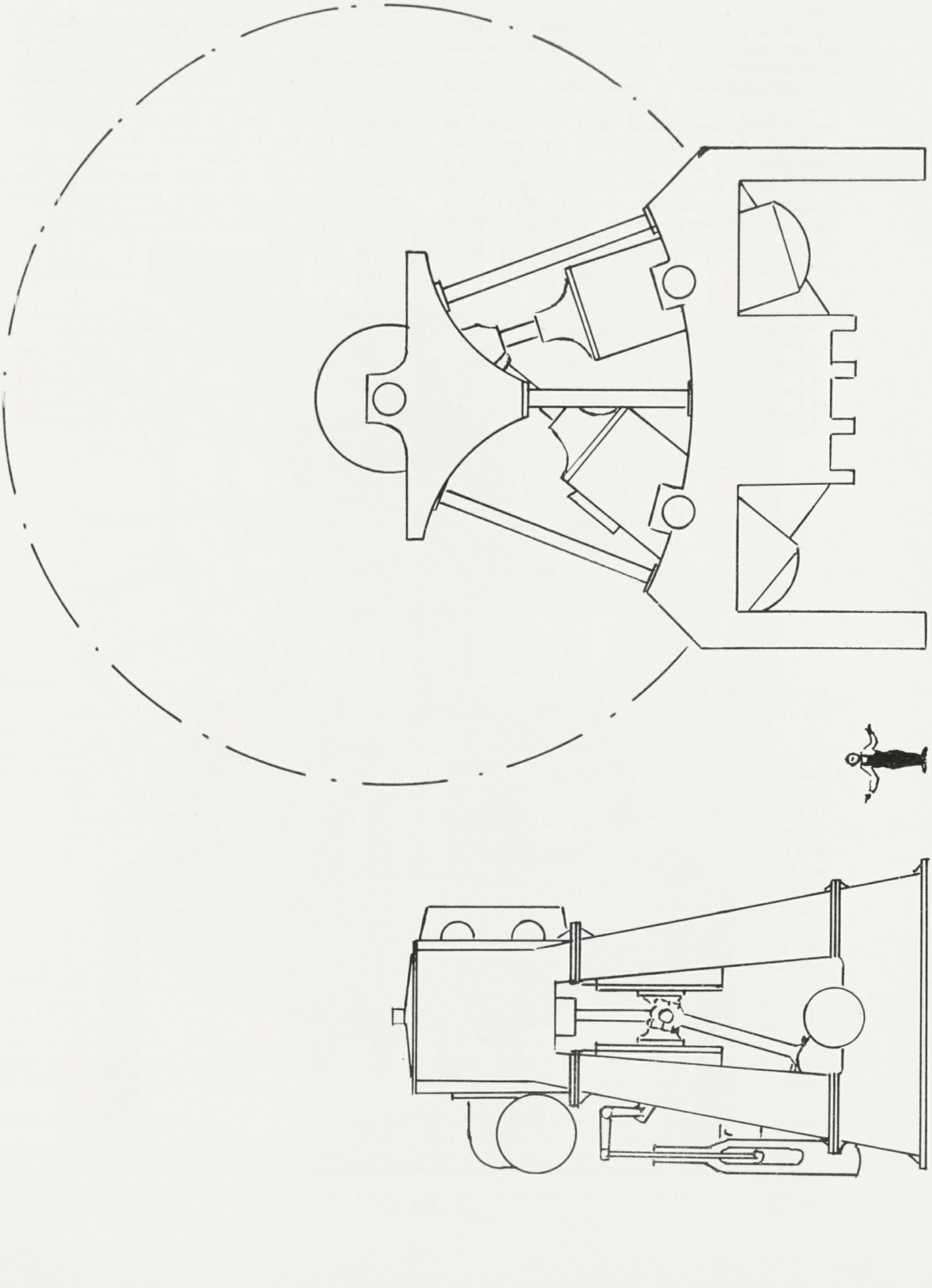


Fig. 1

friction strap the outer end of the crank-pin works and drives the wheels. This friction strap allows the engine to be detached at will from either or both paddles".

J. Bourne states that "In these oscillating engines, the cylinder bottom is cast in (i.e., the cylinder and bottom form an integral casting) whatever be the size of the cylinder". He goes on to say that "the air pumps are wrought from a crank in the intermediate shaft, as is usual in oscillating engines, and there has been great difficulty in getting a sound forging made of such a difficult form and of such large dimensions". In the days of jet condensers, the volumetric ratio of steam cylinder to air pump cylinder was about 7·5 which for this engine gives a swept volume of about 110 cubic feet per pump. This would require a crank throw of some 3 ft., an awkward operation on a 24 in. diameter forging.

He proceeds: "The slide valves are equilibrium gridiron valves, but as a considerable loss of steam would be caused by filling the passages every stroke, if only a single three-ported valve was used, it has been thought advisable to use two valves with two ports in each for each cylinder, whereby this loss is prevented". (Fig. 2.)

Steam from the boilers was delivered to the cylinders through the outer trunnions and exhausted through the inner trunnions to twin condensers.

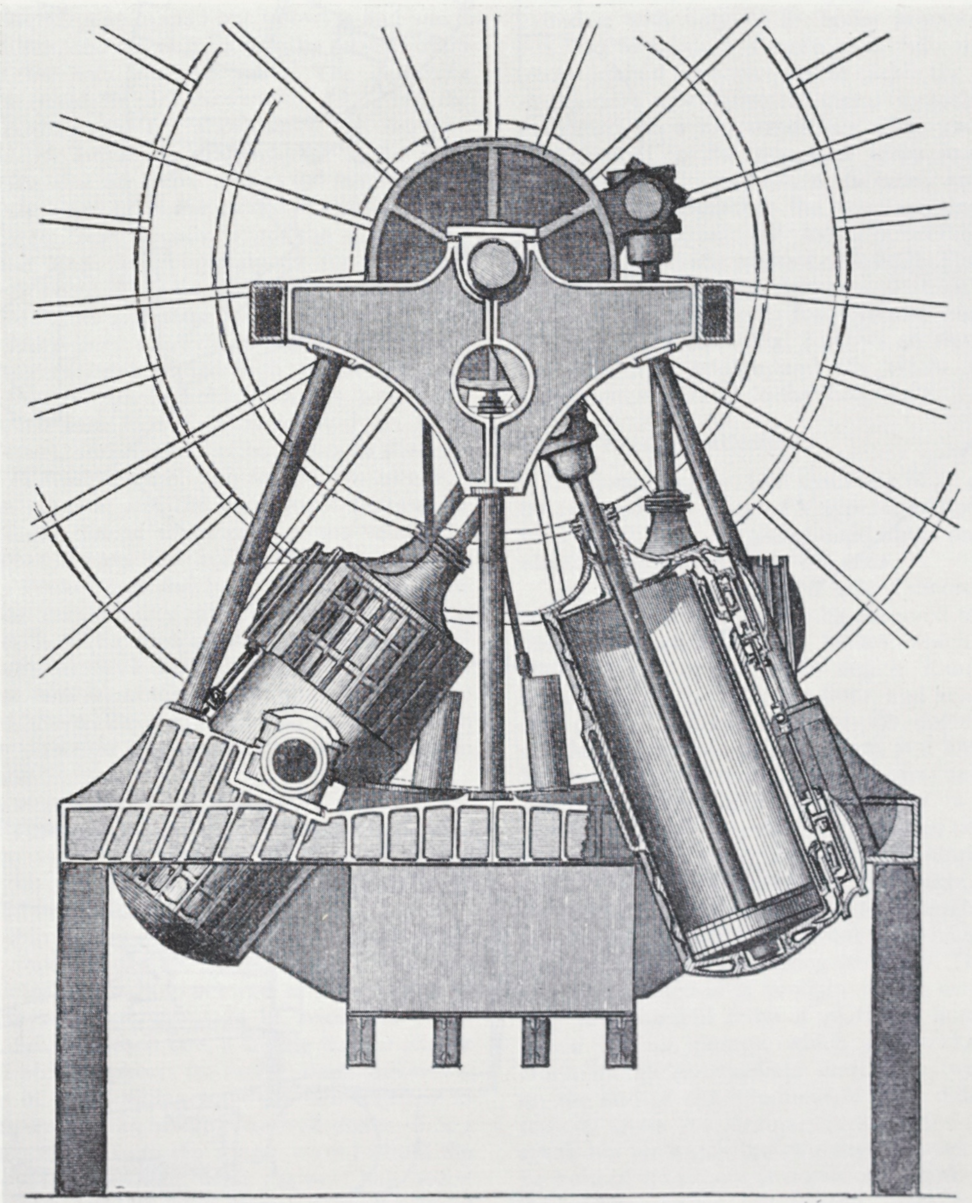


Fig. 2

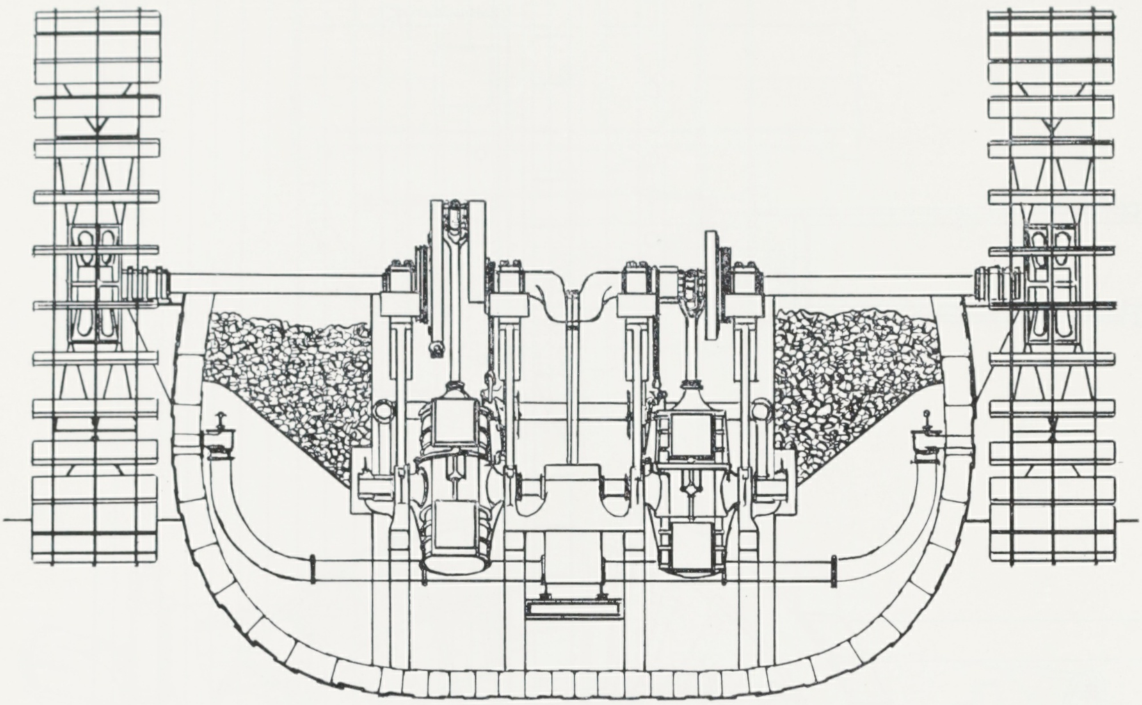


Fig. 3

Paddles

The paddles originally fitted were 56 ft. in diameter, but on the third voyage outward bound, these were both lost in a storm, and the new wheels fitted had a diameter of only 50 ft. Even this diameter seems excessive, if the empirical formula of the day can be relied on:—

$$\frac{\text{Knots X 100}}{\text{r.p.m.}} = \text{Circumference of wheel}$$

The wheels had radial floats, i.e., they were fixed wooden floats as opposed to the cast iron feathering floats worked from an outside eccentric rod or "Jenny Nettle". The reason for such a retrograde step seems to have been the number of failures due to moving floats.

A turning rack was mounted on the circumference of each wheel, as shown on Fig. 4, but this may be considered a refinement as each wheel could be disconnected and turned for inspection by means of a pair of chain blocks.

Screw Engine Particulars

Reciprocating horizontal vis-a-vis engine, with four cylinders 84 in. diameter by 48 in. stroke, developing 4,886 i.h.p. at 38·8 r.p.m. giving nine knots on screw alone.

J. Bourne gives a good general description of the engine, Figs. 5 and 6: "These engines are the largest screw engines hitherto constructed. The combination consists of four cylinders which are laid upon their sides. From each of these cylinders two piston rods proceed to a horizontal crosshead which moves in guides, and from the crosshead of each cylinder, a connecting rod proceeds to the crank. There are two cranks in the shaft, and two of the cylinders operate on the one crank, and the other two cylinders operate upon the other crank. There are two connecting rods therefore acting upon each crank, and the piston of one engine is at the top of its cylinder when the piston of the opposite engine is at the bottom of its cylinder. Each connecting rod consists of two rods. Between the cranks there is a great disc introduced, the purpose of which is to balance the momentum of the moving parts.

"The condensers are situated between the cylinders, and the air pumps are wrought off upon the crossheads. The steam is conducted from the cylinders to the condensers by arched pipes, which are conspicuous in the drawing, and the steam enters from the boiler by the round pipes shown on the tops of the valve casings. The square pipes projecting through the ends of the condensers are the waste water pipes.

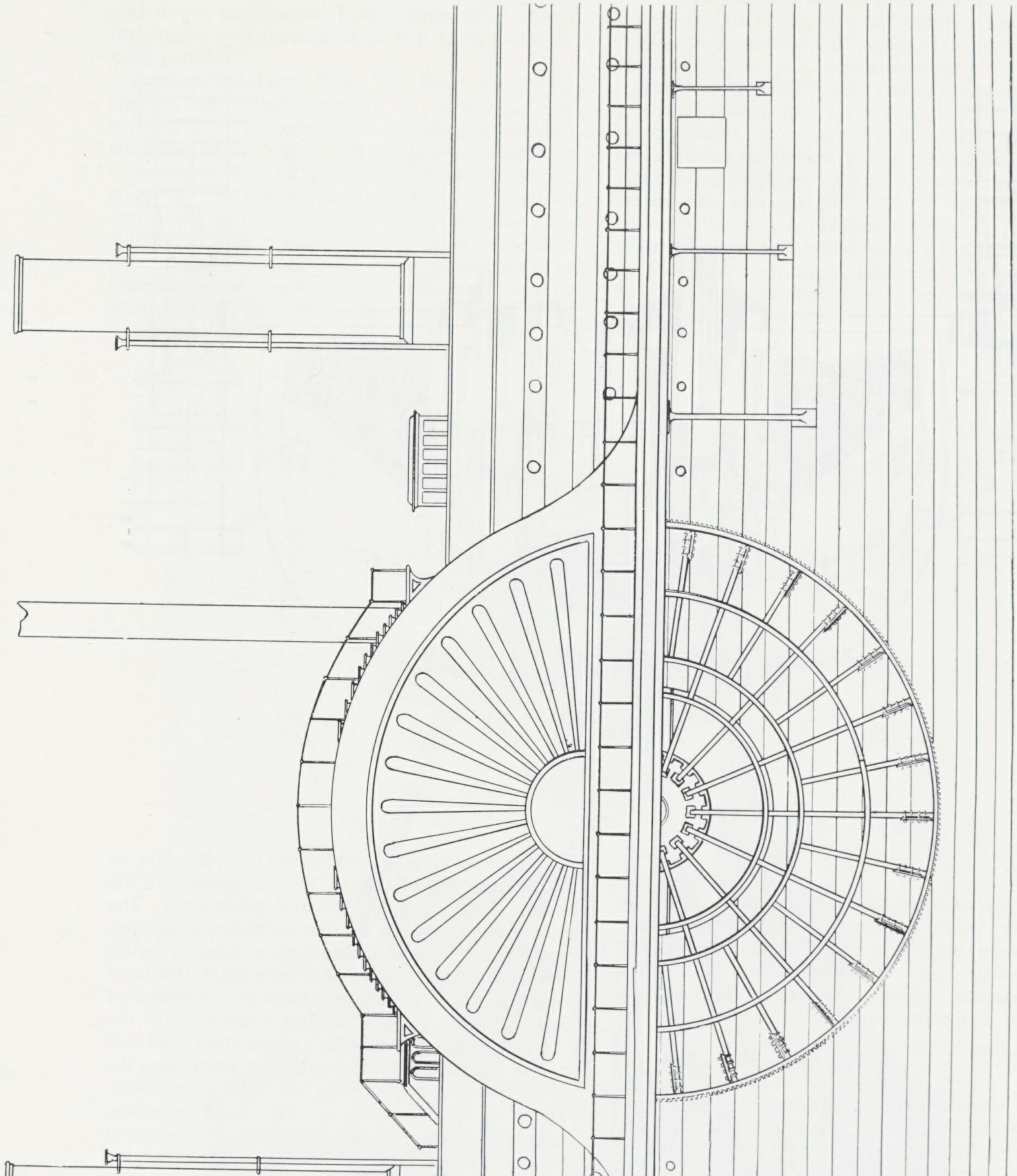


Fig. 4

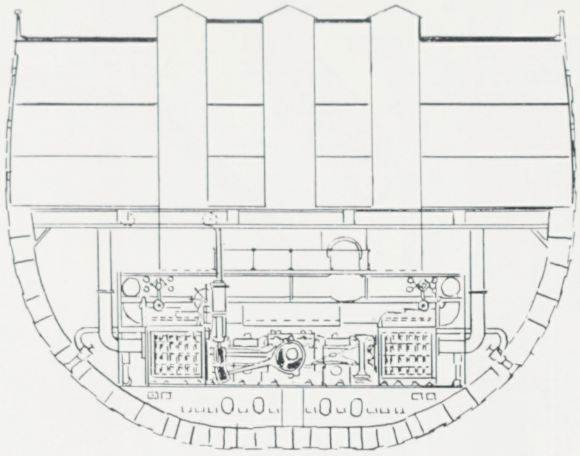


Fig. 5

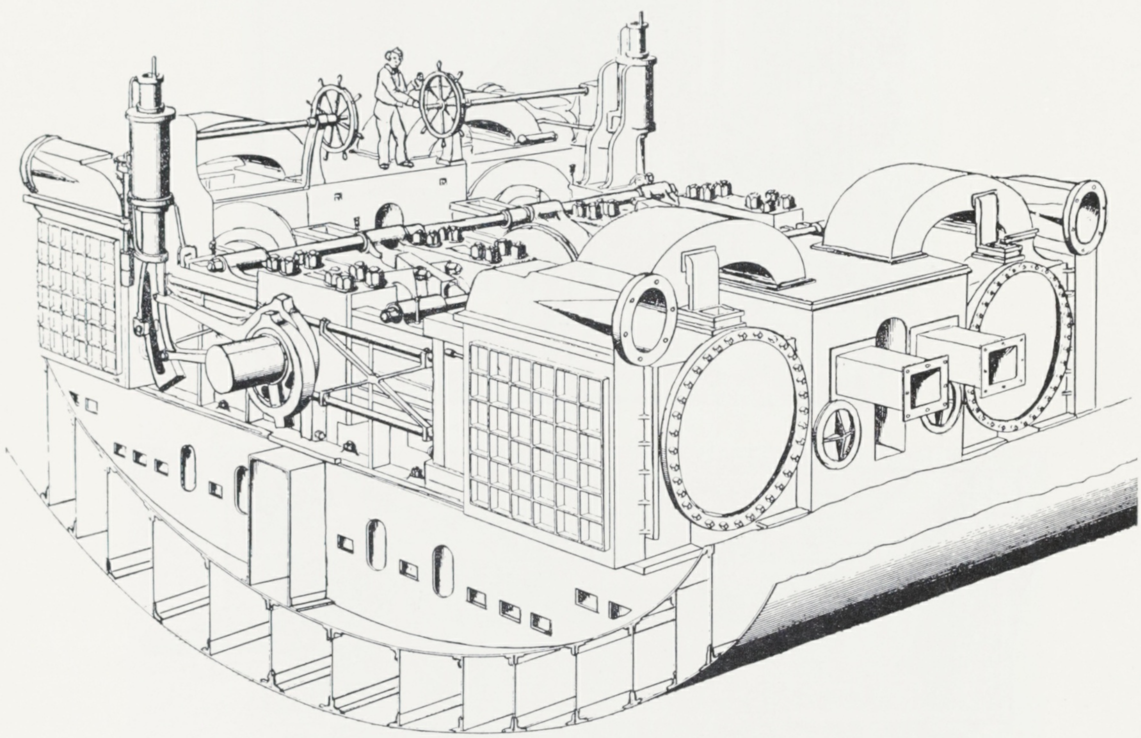


Fig. 6

"The slide valve which works upon the side of each cylinder is moved by a framework shown at the end of the engine, and to which motion is imparted by the link motion. The link of the link motion may be moved up or down, either by a small cylinder placed over each link for that purpose, or by a screw turned by bevel wheels, and to which motion is given by a handwheel moved by the engineer. Of the slide valve, a section is given in Figs. 7 and 8. It is a gridiron valve with a ring at the back to take off the pressure, and the weight of the valve is borne by rollers introduced for that purpose. The back of the valve casing which is removable and which is planed and scraped very true on the inside for the brass ring on the back of the valve to rub against, is formed with a multitude of cross bars on the outside to prevent distortion from the great pressure exerted by the steam over so large a flat surface. Circular doors are placed on the ends of the condensers opposite to each air pump, by which access to the air pump valves may be obtained and through which the bucket may be withdrawn."

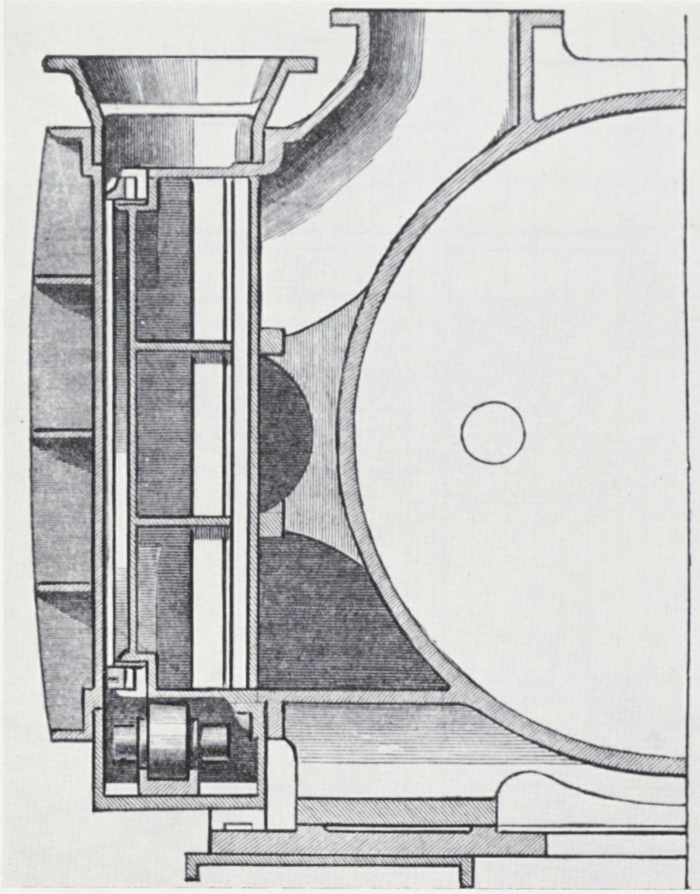


Fig. 7

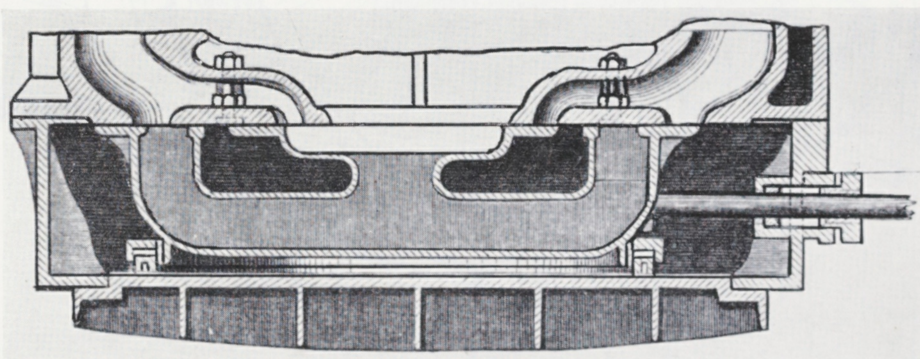


Fig. 8

Fig. 9 shows details of the piston with its double rods secured by round nuts let into the body, while Fig. 10 shows the crosshead with the bracket for driving the air pump.

J. Bourne continues: "Fig. 11 shows side and edge views of the connecting rod. The cylinders upon the one side of the vessel (starboard side) have each one connecting rod of this kind, and the cylinders upon the other side of the vessel have each two connecting rods of the same description but of lighter make. The object of this arrangement is to enable the opposite cylinders to be kept in the same vertical plane".

Mr. Bourne seems to have anticipated Sir Charles Parsons' arrangement of high pressure reciprocating engines exhausting into low pressure

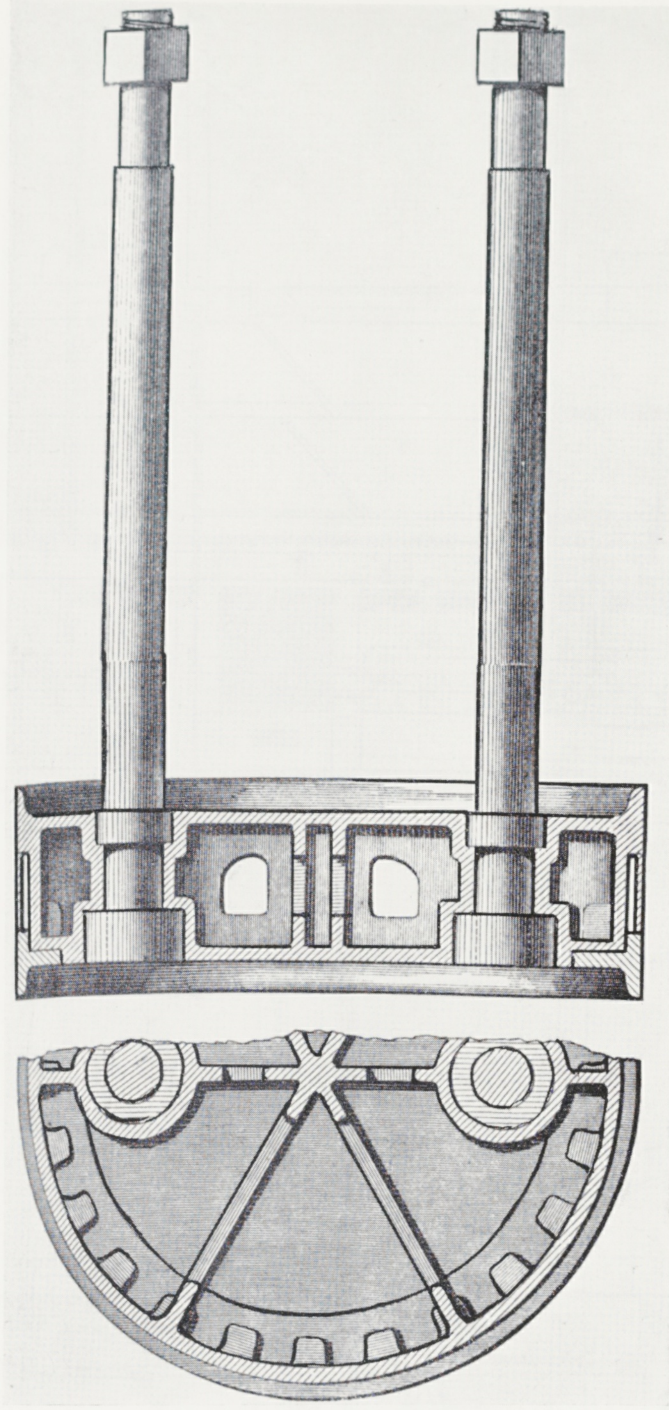
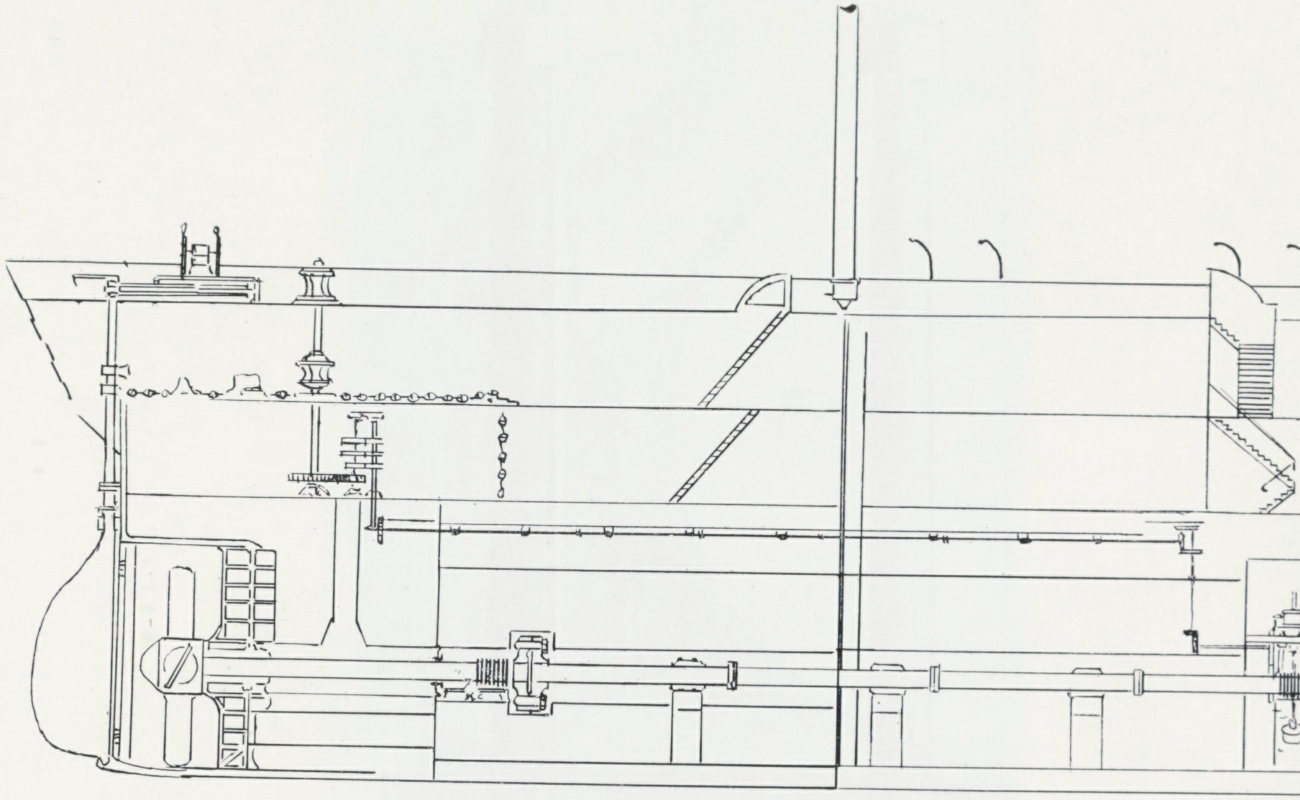
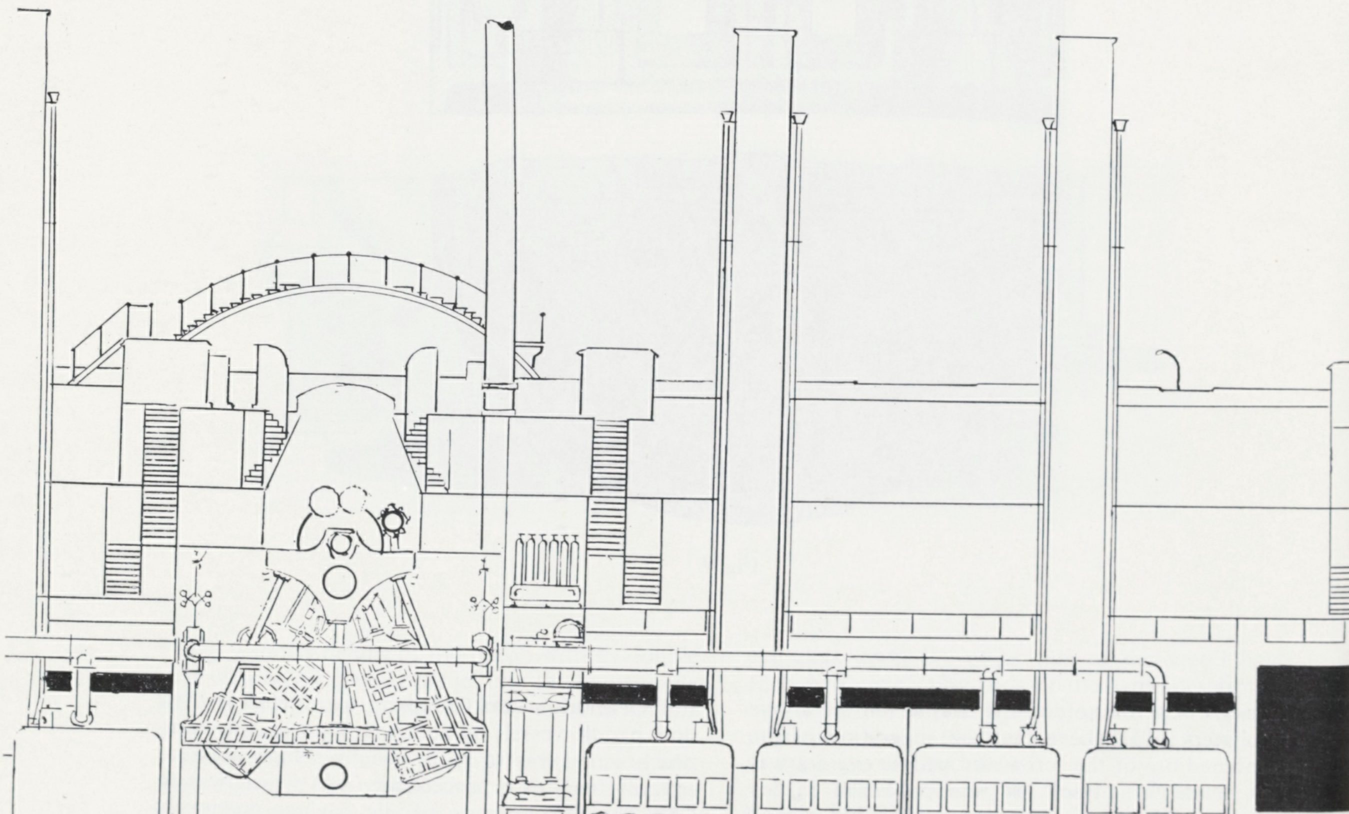
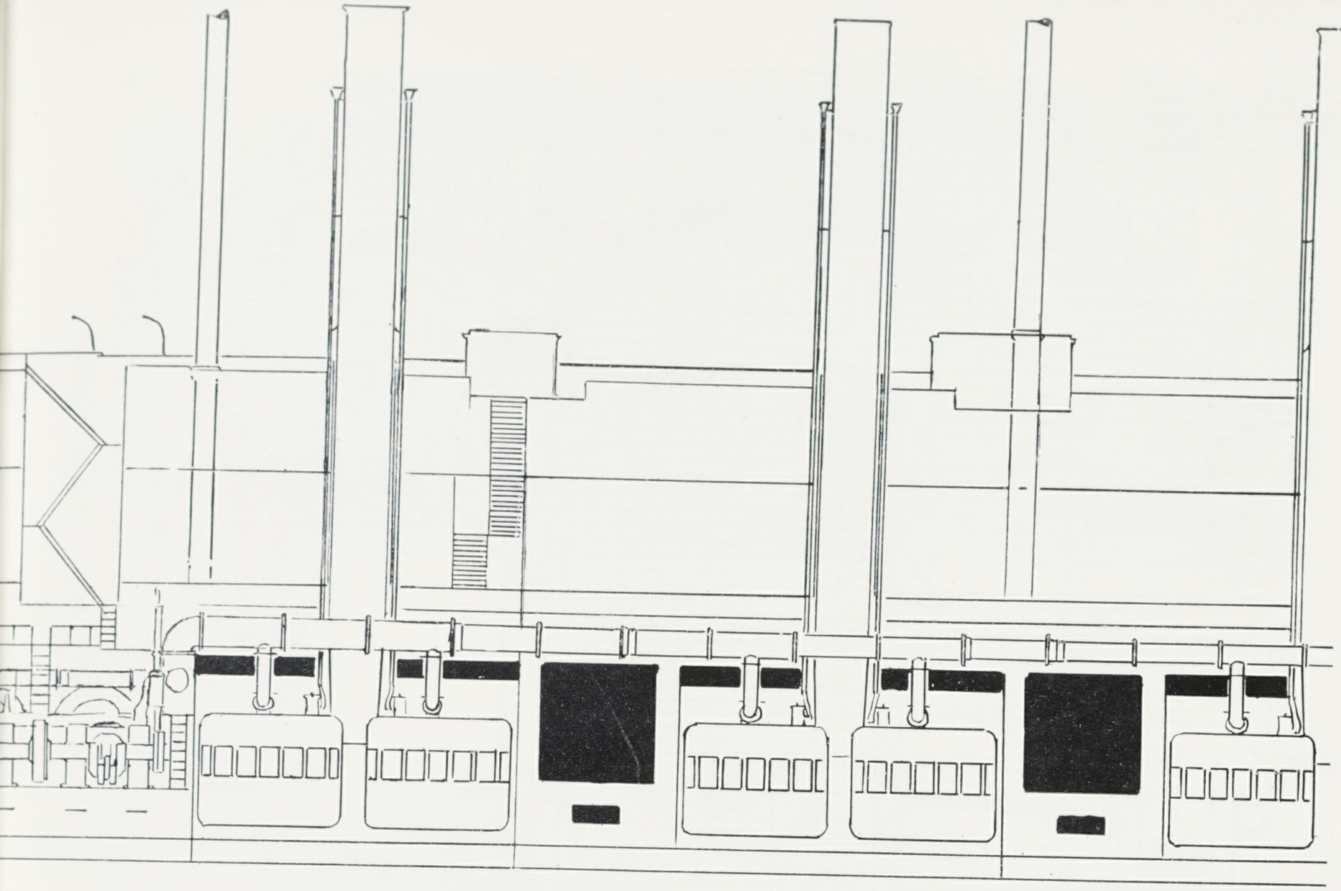


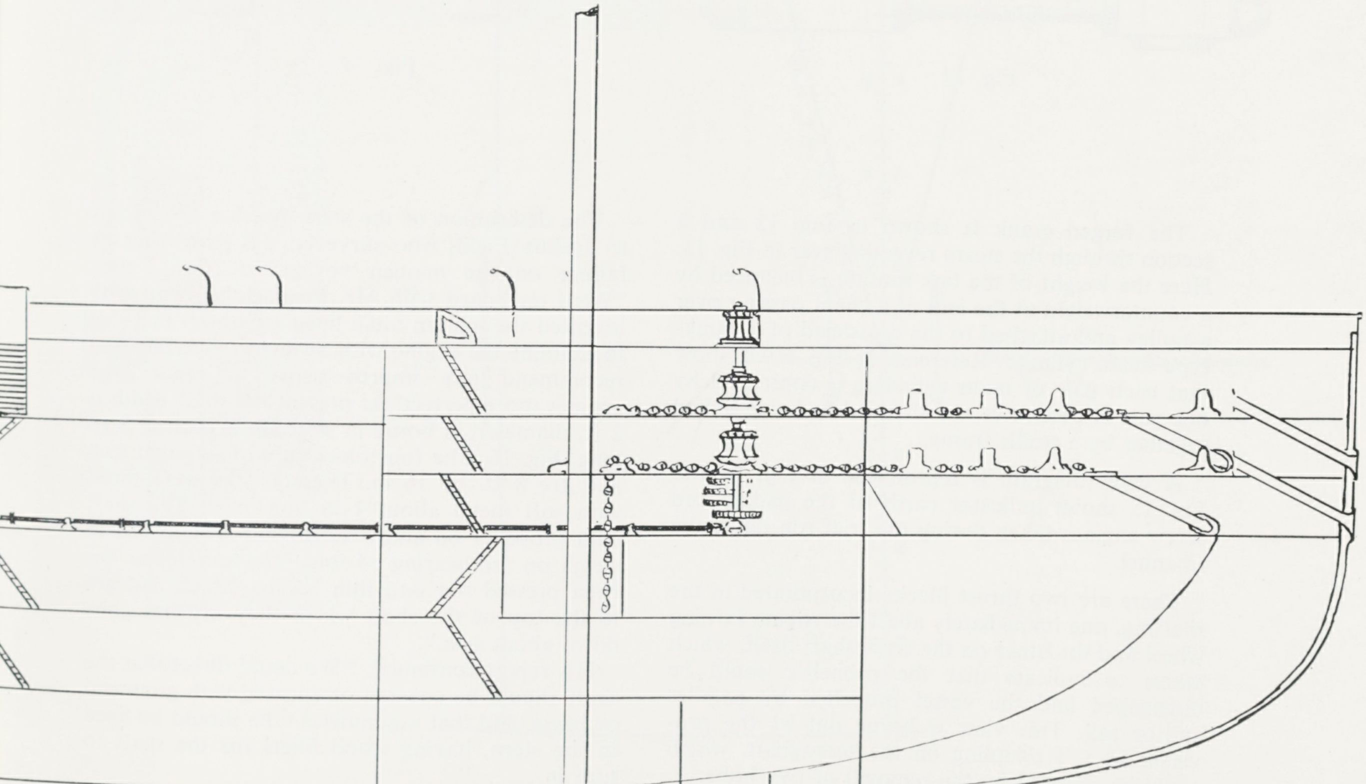
Fig. 9

turbines in a footnote to his chapter on the above machinery: "I believe that the suggestion of the combined use of the screw and paddle engines was first made by myself. It was published in my treatise on the screw propeller in 1851, but several years before that I recommended the Peninsular and Oriental Steam Company to accelerate their slow paddle vessels by adding a screw in the stern, which screw was to be driven by a high pressure engine—the steam proceeding from which was to drive the low pressure paddle engines previously in use".









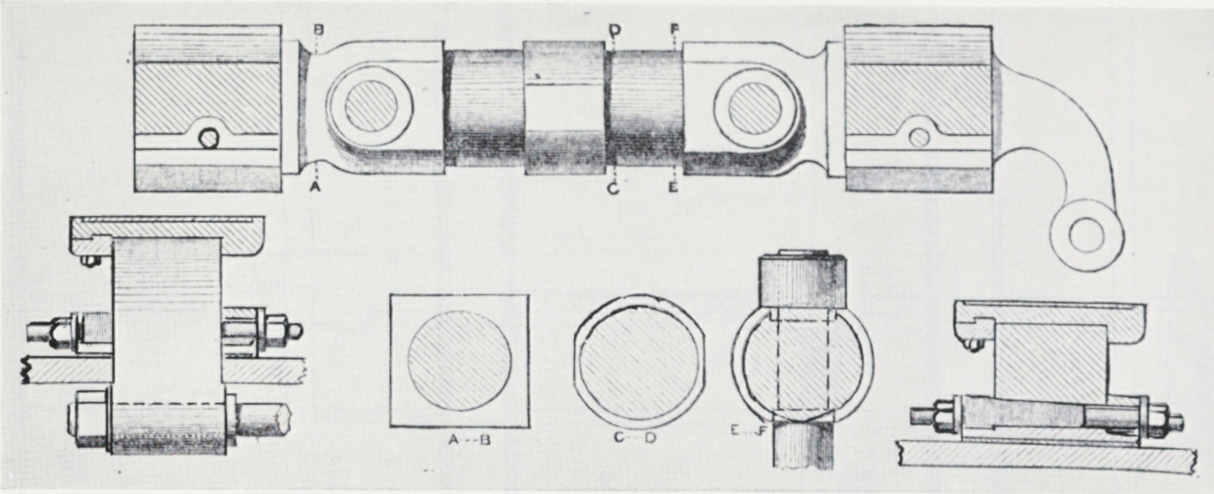


Fig. 10

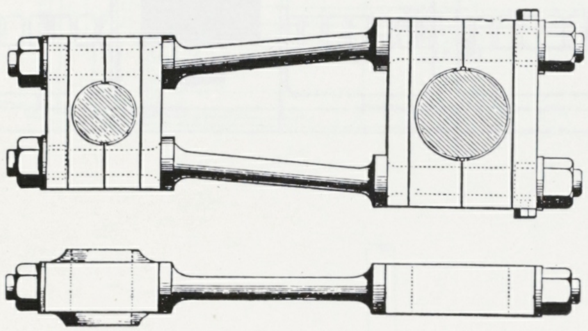


Fig. 11

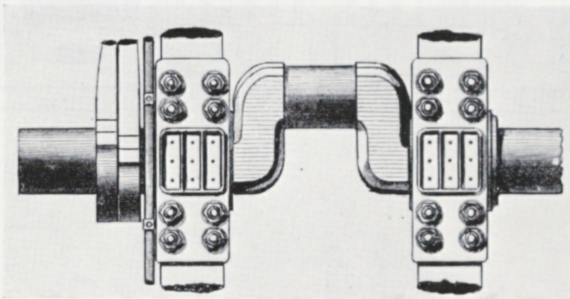


Fig. 12

The forged crank is shown in Fig. 12 and a section through the steam reversing gear in Fig. 13. Here the weight of the link motion is balanced by a counterweight at the end of a chain passing over a pulley and attached to the crosshead of a trunk-type steam cylinder. Reference to Fig. 6 will show that each pair of main cylinders is controlled by one pair of eccentrics, the slide valves being linked together by a trellis frame.

A valve diagram is reproduced in Fig. 14 and Fig. 15 shows indicator cards of the paddle and screw engines taken during the trial run down the Channel.

There are two thrust blocks incorporated in the shafting, one immediately abaft the engine turning wheel and the other on the screwshaft itself, which seems to indicate that the propeller could be disengaged and the vessel propelled by paddles and/or sail. This view is borne out by the provision of a T coupling on the screwshaft, which could be released by the removal of two bolts.

The description of the stern bearing is best left to Joshua Field, who surveyed this item after its failure on the maiden voyage to New York: "Went on board with Mr. Penn (John Penn who invented the lignum vitae lined sternbush in 1854) to examine the engine with a view to ascertain and recommend any improvements to render the engines more perfect. At present the shaft width is 2 ft. diameter, is borne in stern, in a bearing 8 ft. long, Fig. 16. The four blocks are of wrought iron and are 8 ft. by 16 in. These blocks were faced with soft metal about ½ in. thickness. The shaft with propeller on and overhanging is supposed to weigh on the bearing 54 tons. The soft metal has been pressed out into thin lamina which had got to the top of the shaft which they say has gone down about ½ in.".

His report continues: "We could direct that the shaft should be covered or encased with gunmetal or brass, and that a gunmetal tube should be fixed in the stern, having wood fillets for the shaft to turn in . . . .".

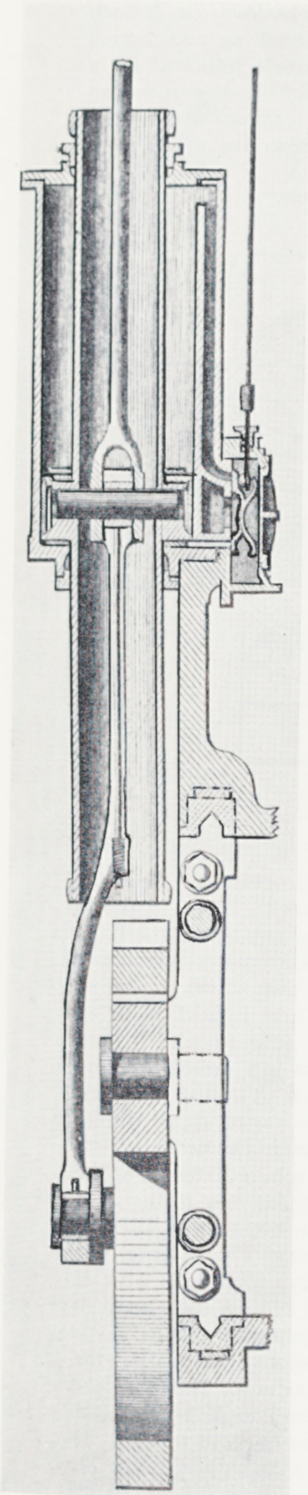


Fig. 13

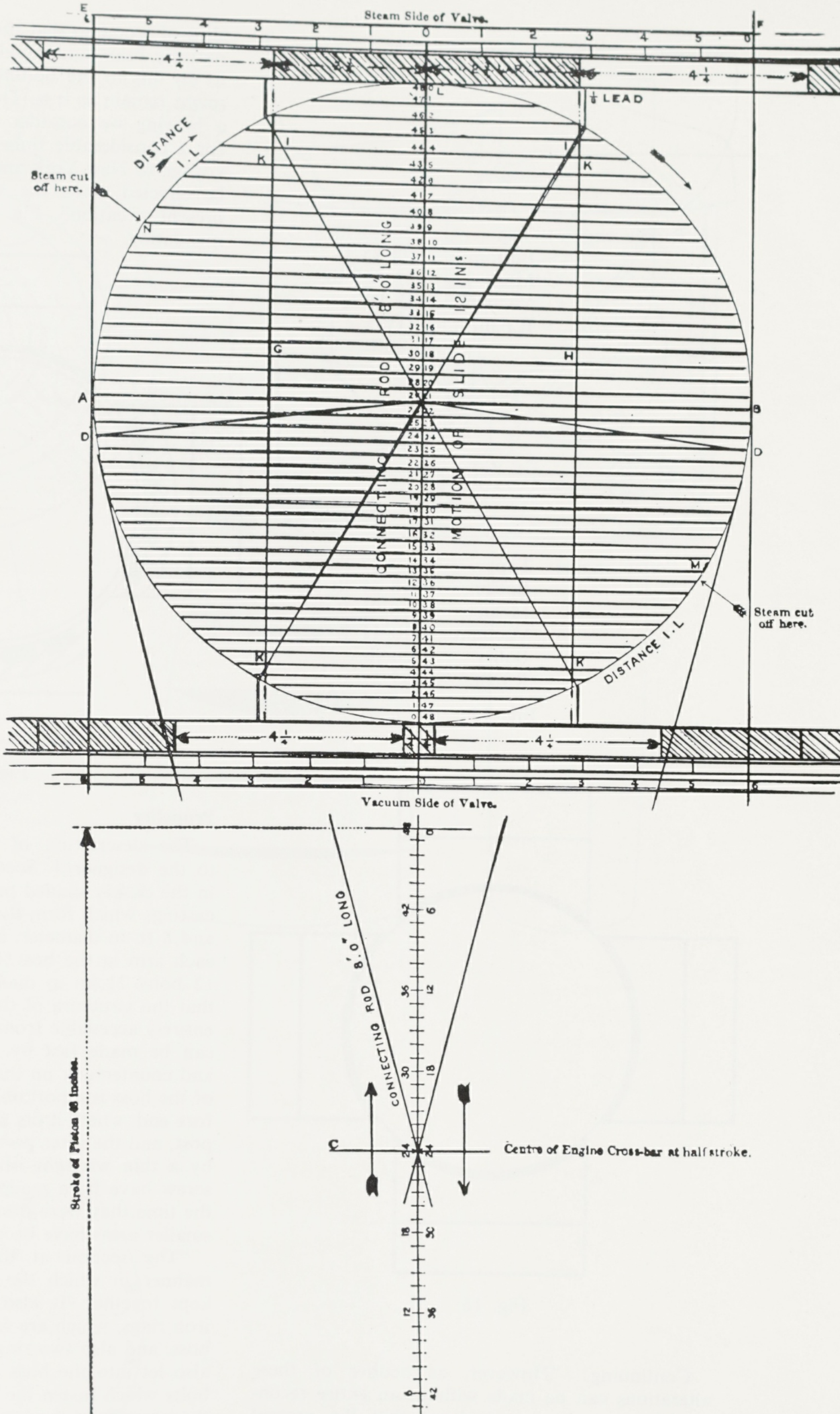
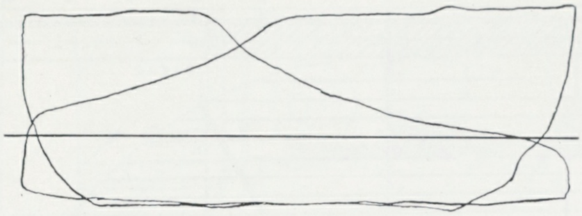
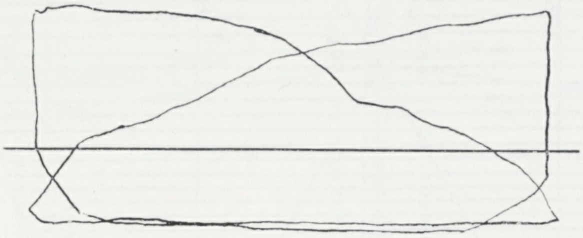

Steam Side of Valve.
¼ LEAD
DISTANCE I.L
Steam cut off here.
CONNECTING ROD 8' 0" LONG
MOTION OF SLIDE 12 INS
Steam cut off here.
DISTANCE I.L
Vacuum Side of Valve.
Stroke of Piston 48 Inches.
CONNECTING ROD 8' 0" LONG
Centre of Engine Cross-bar at half stroke.

Fig. 14



Paddle engine: Top 20·40 lb.
Bottom 24·85 lb.
Mean 22·625 lb.
Vacuum 12 lb.
R.p.m. 11·75



Screw engine: Top 19·27 lb.
Bottom 18·55 lb.
Mean 18·91 lb.
Vacuum 10–11 lb.
Boiler 20 lb.
R.p.m. 35·5

Fig. 15

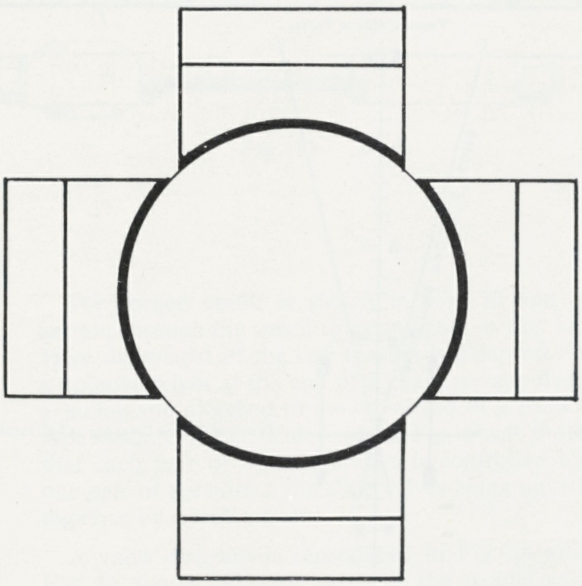


Fig. 16

Continuing: "However, as neither of these alterations can be made without an entire reconstruction, we have considered that the present wrought iron blocks which contain the white metal be planed away to the extent of 2 in., and a thick brass or gunmetal bearing fixed upon these slabs to have dovetail grooves cut in them, to receive slips of wood so that the bearing should be alternatively brass and wood. This would be sufficient to do this to the bottom and two sides. The top might remain as it is. There is no wear on it. Such a bearing we consider will work so constructed for a considerable time and quite sufficient for a voyage to New York and back. The operation can be effected while the ship is afloat and in her present situation". Fig. 17 illustrates this recommendation.

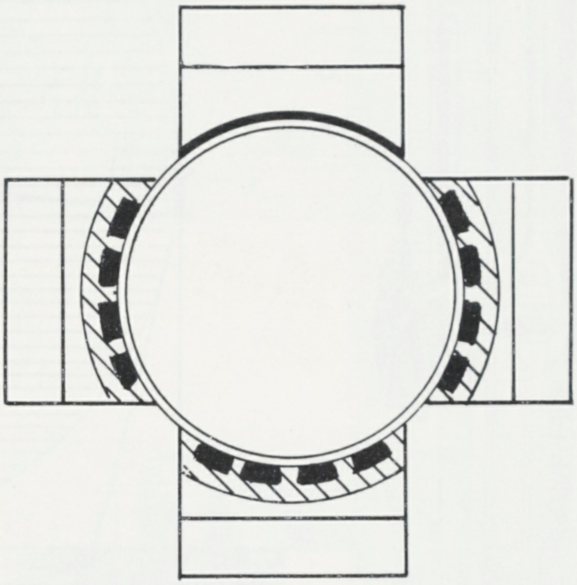


Fig. 17

Propeller

The description of the propeller is best left to the designer J. Scott Russell: "Fig. 18 shows in the darkly shaded part, the solidity of the large castings which form the boss, which is cylindrical, and 8 ft. in diameter, and it also shows the set of each arm in the boss. Each arm is held down by 12 bolts 2½ in. in diameter. It is to be observed that the structure of the boss is a hollow casting, entirely accessible from within, so that all the bolts can be made fast by nuts screwed from within, and countersunk on the outside. The general form of the boss is a portion of a sphere flattened at the fore end, where it fits the circular boss of the stern post, and the after part of the screw boss is closed by a thin wrought iron casing. The arms of the screw have been made smaller than they were at the time they were designed, but since then, much smaller arms have become generally used.

"The section at the bottom left shows the manner in which the two sides of the boss are kept together. It also shows two large wrought iron rings, which are fastened on both sides of the boss, and also two rings on each side of the joint, also let into the boss of the screw, to aid the 28 bolts which fasten the sides together in preserving the integrity of the boss."

The propeller diameter was 24 ft., its pitch 37 ft. and it weighed 36 tons. The illustration shows that the propeller was hardened up on to the shaft by means of two cotters.

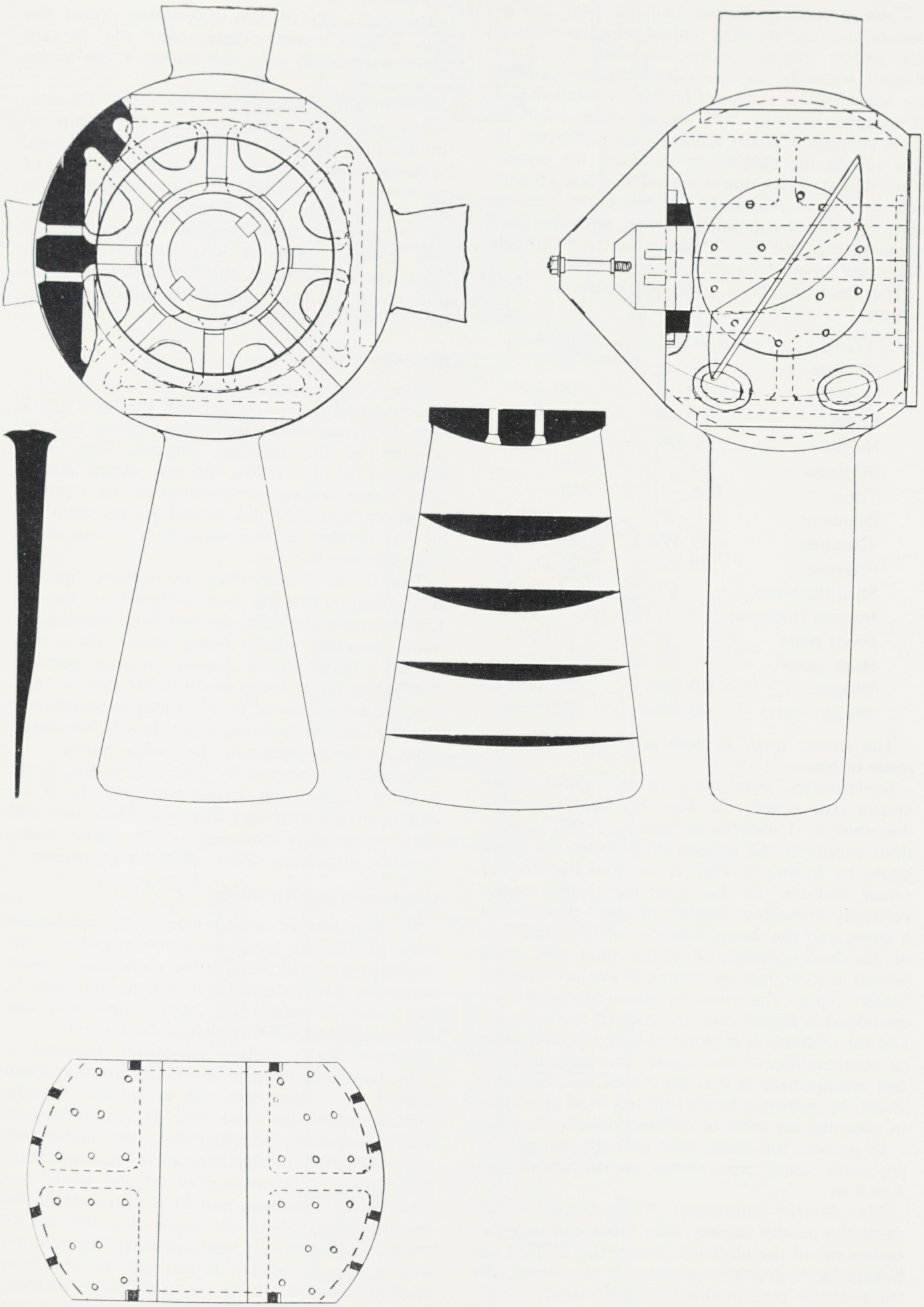


Fig. 18

Boilers

Each propelling engine had its own set of boilers, the four forward boilers being specifically the paddle engine boilers, supplied by Scott Russell, while the six after ones were provided for the screw engine and were built by J. Watt & Co. If required, however, both sets of boilers could be cross connected. Apart from a slight difference in size and in the number of furnaces, the boilers were similar, being double-ended and box shaped, and they were stayed in all directions. Fig. 19 gives a view of the paddle boiler, with five furnaces a side, and Fig. 20 shows sections through the screw boiler, with six furnaces a side.

The following table gives particulars of both boilers: —

ITEM	PADDLE	SCREW
Number	4	6
Length	17′	18′ 4¾″
Width	17′ 9″	17′ 6″
Height	13′ 9″	14′
Furnaces	10	12
Tubes	800	840
Diameter	3″	3″
Thickness	12 WG	10 WG
Pressure	25 lb.	25 lb.
Shell thickness	⅜″	7/16″
Bottom thickness	7/16″	½″
Front plate	½″	½″
Back plate	9/16″	⅝″
Weight	40 tons	55 tons
Weight water	40 tons	45 tons

The smoke tubes in both sets of boilers were made of brass.

Superheaters were fitted to the screw engine boilers (for details *see* Fig. 21), and these are described by J. Bourne as follows: "The illustrations represent the superheating apparatus introduced by Messrs. J. Watt & Co. into the steamer *Great Eastern*. In this case the smoke passes vertically through a number of small pipes set in a chest, and the steam which is let in at one side of the chest is drawn off at the other side, being heated in its transit by coming in contact with the heated pipes. The chest in which the pipes are contained is placed over the root of the chimney, and the chimney is in point of fact a prolongation or continuation of that chest. One advantage of this arrangement is that the tubes may be easily swept, by passing a brush through them as is done in sweeping the tubes of ordinary tubular boilers".

In passing, the reader may note the diameter of the main steam pipe, shown on the drawing as 3 ft. 9 in.

Mr. Bourne continues: "The boilers which drive the paddle engines were fitted with a water casing round the chimney, which was to feed the boilers by hydrostatic pressure of the water. To the overflow pipe of this casing, however, a cock was injudiciously affixed, which cock having been incautiously shut, the casing was burst on the first trial trip by the accumulated pressure of the water and steam within the casing, causing much alarm and injury".

The explosion of this economiser killed five men, injured many others, blew the forward funnel out of the ship and started a fire in the main saloon.

One of the proposals for the *Great Eastern's* boilers seems to have been an early form of thimble-tube type (*see* Fig. 22). The thimbles could be easily removed for access to the water space by unscrewing the bolt, and the latter served also as a stay to keep the sides of the firebox from bulging. Apparently these thimble tube boilers had been used in America long before the *Great Eastern* was thought of, and had been used in this country at the beginning of the nineteenth century by Beale and Barrans.

Auxiliary Machinery

Comparing the engine room of to-day with that of a hundred years ago, perhaps the most significant difference would be in the number and disposition of the auxiliary engines. The hardy passenger of the 1860's did not expect electric light, steam heating, air conditioning or a private bathroom, and if he did, he did not get them. All he was entitled to was something to eat and a place to sleep in.

The *Great Eastern* had no electric light or steam heating, having been designed for the Far Eastern trade. As built, she had hand steering, the double steering wheels being placed aft in way of the rudder head. Steam steering gear of McFarlane Gray design was installed only in 1867. She had no system of engine room telegraph, and steering and manœuvring orders had to be relayed from the bridge spanning the paddle boxes.

As will be observed by reference to Fig. 3, the forward engine room casing was made to fit the paddle engine with very little space left over, and the only ancillary machinery in the engine rooms were the air pumps driven off the main engines.

Condensers and Air Pumps

In both sets of main engines, the condensers were of the jet type, there being two for the paddle engine and four for the screw engine. Each condenser had its own air pump placed inside it, and these air pumps did duty as circulating and extraction pumps, sometimes as feed pumps.

Samuel Hall's surface condenser, patented in 1834, had been fitted to several steamers and had proved very successful but it did not become popular until the 1860–70's. The objections to Hall's condensers were that they were unnecessary and expensive, and that they soon became clogged with grease. A great deal of oil and tallow was used in the cylinders, and in a statement of the relative merits of the jet condensing and surface condensing engines, it was said that: "In injection engines the oil which is put into the cylinders, stuffing boxes, slides, etc., is speedily carried away by the injection water into the sea, while in the surface condensing engines, not a particle of the oil which is given to the internal parts of the

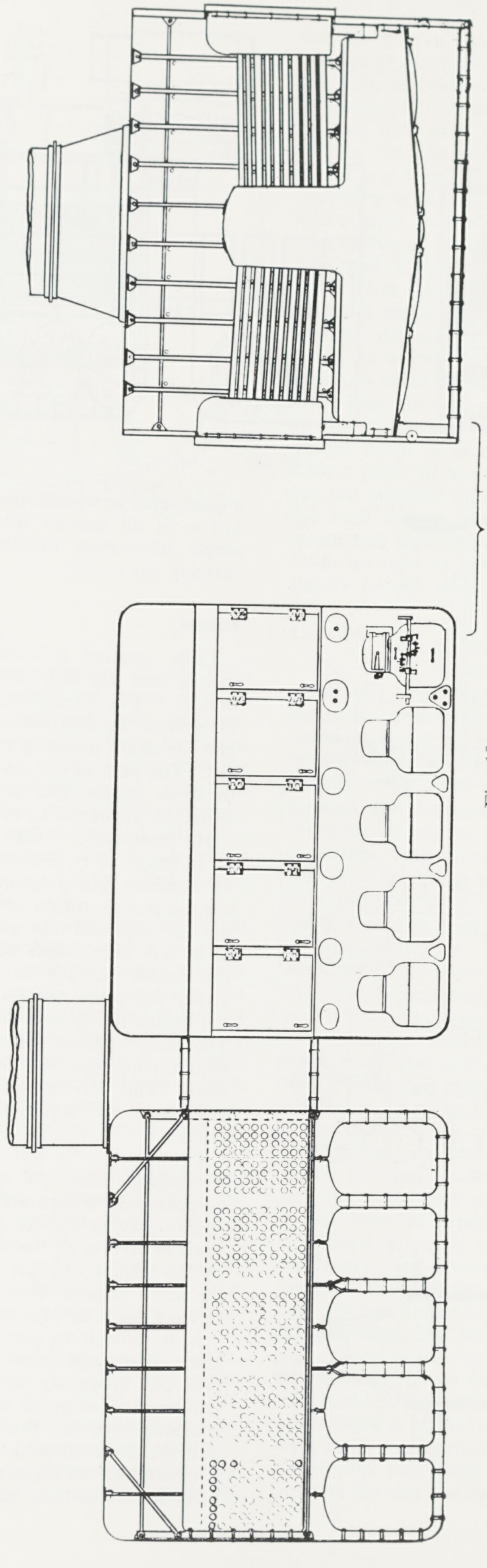


Fig. 19

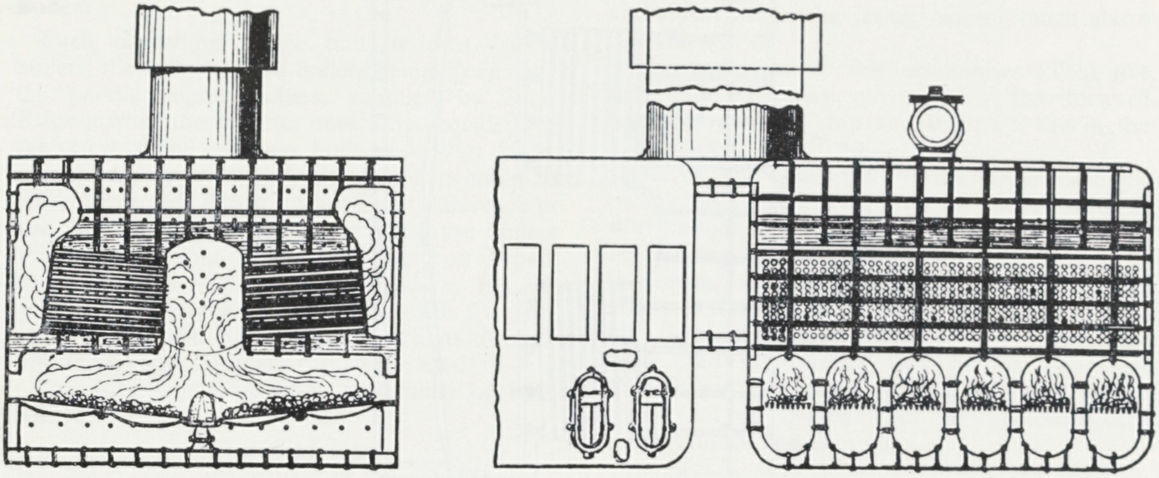


Fig. 20

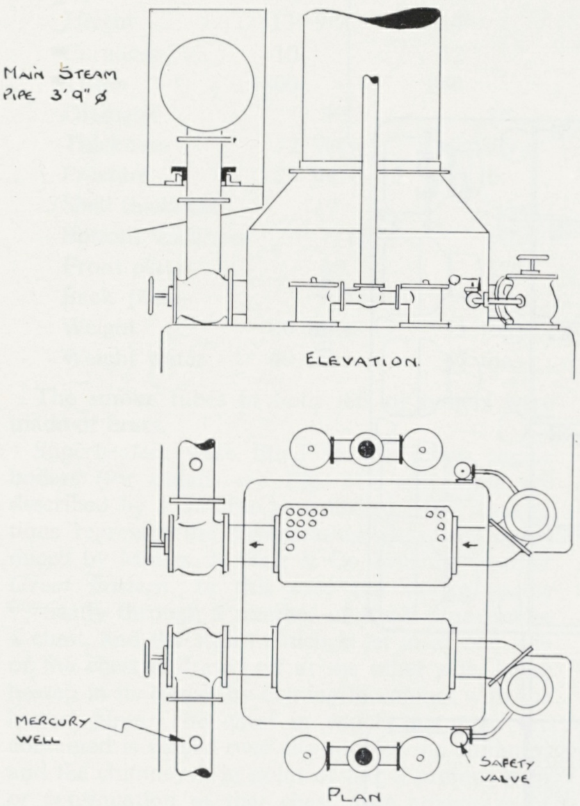


Fig. 21

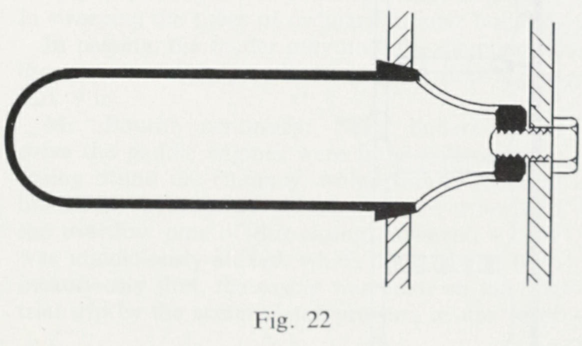


Fig. 22

engine, etc., is wasted away into the sea, or lost, but it is all carried into the boilers, whereby ample lubrication of the engine is effected at scarcely any cost".

Pumps

Little mention is made of the feed arrangements, and the only clue is that given by the builder: "Fig. 23 shows the general arrangement of boilers, the distribution of fuel all round them and the steam pumping engines by which they are fed". The plan of the machinery spaces shows ten of these pumps, one for each boiler, and they would all appear to be of the banjo type.

He continues: "The paddle engine room is 40 ft. long: 10 ft. are reserved in front of this for an auxiliary pumping and working engine. Here are the pumps, meant in case of accident to pump out the ship, and they are worked by two engines of 40 h.p. each. These pumps also assist to empty the double skin of its water ballast; and provision was made there also, if necessary, to light the engine rooms and the ship generally with gas. The same engines, were, also, made to connect with driving shafts to work the capstans of the ship". (This driving shaft was about 200 ft. long and ran part of the way through the accommodation.)

Continuing: "It will also be noticed here, that immediately abaft the screw engine room a small compartment is separated, containing an auxiliary engine, having for its object to cause the screw shaft to revolve in case of the ship being propelled by the paddle engines, without the assistance of the screw engines. This engine might also be used to work the after capstan".

It is difficult to accept this statement without reservation; the screw engine was after all the principal propelling agent, and if it had to be temporarily stopped, the two bolts could be removed from the T coupling on the screw shaft and the latter allowed to trail. Otherwise, why fit a thrustblock on the propeller shaft? Possibly this auxiliary engine was intended as a turning engine.

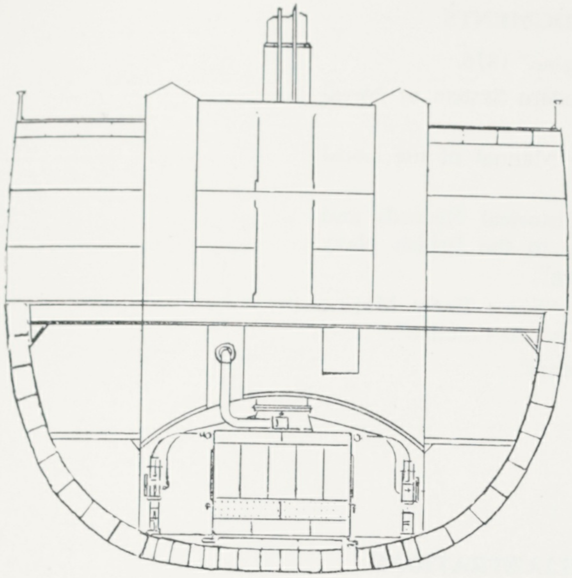


Fig. 23

Pumping Arrangements

The jet condensers were supplied directly from the sea through a Kingston non-return shut off valve on the skin of the ship, and a stop cock controlled from the manœuvring platform, by means of which the amount of water injected into the condenser could be regulated—hence the term "Main Injection". Another cock on the condenser was fitted with a pipe and strum and led to the bilge, whereby the condenser and its air pump could be used as an emergency bilge pump—hence the term "Bilge Injection".

Opening and closing the injection cocks on starting and stopping the engine was a very important duty, and it was a simple enough mistake completely to flood the engine with sea water. On starting it was absolutely necessary first to form a vacuum in the condenser, and this was done by flooding the engine with steam until it was seen coming from the snifting valves. This having been done, the injection water was turned on, a partial vacuum formed and the engine could then be moved.

Steam cylinders were lubricated with tallow, and tallow injectors were a standard fitting on most boilers to prevent priming. What with the tallow injected into the boiler added to that brought in through Hall's patent surface condenser, it is small wonder that boiler explosions came to be considered by the marine engineer as an occupational hazard.

Jacob Perkins, pioneer of high pressure steam, voices the philosophy of the day in his description of the new high pressure boiler: "High pressure engines built on Oliver Evan's plan are known to be perfectly safe, owing to their peculiar construction. The boiler, which is considered to be the only dangerous part of a steam engine, when made on the Oliver Evan's plan never explodes. The boiler consists of a cylinder made of wrought sheet iron of $\frac{5}{16}$ in. in thickness, its diameter is 30 in. and its length 18 ft. At each end of the cylinder a disc of cast iron of 4 in. in thickness is so firmly secured that the wrought iron part of the boiler would give way with a quarter part of the power necessary to explode the cast iron ends. When the boiler gives way, which must necessarily be the case sooner or later, it will be at the weakest part. The rend or opening, which takes place when the boiler gives way to relieve the pressure, discharges no more steam than is usual for the safety valve to emit. An account of upwards of 600 of these rends have been recorded without the slightest injurious effect being produced. The boilers, when from long use they give way, are readily repaired in the following manner: After the boiler is cooled down, a workman enters the main hole with a patch of lead (which is always kept ready for the purpose) and secures it immediately over the rend with a wooden brace. The pressure of steam will close this patch of lead and make the boiler as tight as before. I have known this temporary remedy stand for months".

Conclusion

The *Great Eastern's* failure as a passenger vessel was primarily due to the inability of her successive owners to recognise her worth in the trade for which she was designed, and this fact is bitterly summed up by the builder, J. Scott Russell: "The fuel shown in this plan (section through the bunkers) amounts to 12,000 tons. Working the engines at high pressure, with great expansion, that quantity of fuel would carry the ship once round the world; but as yet the engines have not been worked in the manner intended, it being unsafe to trust the owners with a mode of working requiring so much skill. Hitherto, therefore, the engines have been worked without their proper degree of expansion, and without the economical development of their fuel and power. When the ship shall fall into the hands of skilful owners and managers, there will be no difficulty, with a much less expenditure of fuel, in maintaining the 14 knots an hour for which the ship was designed and which, with a much larger expenditure of fuel, she has already performed; but it would be unsafe to trust such skilful working to any but owners and managers who have shown by a long term of capacity and ability their trustworthiness for such work. All improvements in machinery, and especially in steam navigation, become wise or foolish in proportion to the capacity and ability of the persons to whom they are entrusted. But with the exception of a few able officers of the ship, the whole management of the undertaking has never yet been in the hands that ought to be entrusted with the arduous duty of getting out of that ship the utmost she is able, with good management, in every department to accomplish, and which if she be not destroyed, I am sure she will one day achieve".

ACKNOWLEDGMENTS

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Birmingham Reference Library.

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Session 1961 - 62

Paper No. 2

Discussion

on

Mr. J. Guthrie's Paper

THE MACHINERY OF THE "GREAT EASTERN"

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

Discussion on

Mr. J. Guthrie's Paper

The Machinery of the "Great Eastern"

Mr. H. B. Siggers

I think it would be reasonable to say that the Transactions of the Staff Association would hardly be complete without some record of that extraordinary ship, the *Great Eastern*, and it is therefore fortunate for us that one of Mr. Guthrie's hobbies is an absorbing interest in unusual types of machinery, which has resulted in this paper.

I must confess that, when he first offered to write it, I felt that, while it would certainly interest some of us, it might have only a limited appeal and I am happy to see from the attendance to-night that I was quite wrong.

As Mr. Guthrie tells us, the ship was propelled by sails, oscillating engines and reciprocating engines and it seems probable that, had the steam turbine and diesel engine been invented at the time of her construction, the designers would have found room for them as well.

Fig. 1 shows a comparison in size between the oscillating engines of the *Great Eastern* and the quadruple expansion engines of the *Olympic*. During my apprenticeship days at Southampton, Harland and Wolff did the White Star repairs and I remember one of their apprentices saying that it took about three of them to lift one of the bottom end nuts of the *Olympic* and, considering the enormous size and weight of many of the reciprocating parts of the *Great Eastern*, one wonders how these were handled in the event of any repairs being necessary at sea.

It is not quite clear to me from the illustrations whether the reciprocating engine crankshaft was a single solid forging or was built up from a number of component parts and it would be of interest if Mr. Guthrie could tell us a little more about this.

It appears from the longitudinal view that there was no seal at the after end of the stern tube and that the screw shaft ran entirely immersed in seawater—not a very satisfactory arrangement by modern standards.

Since the ship carried 12,000 tons of coal for bunkers, it is not surprising that there was little space for pay-load and I would ask Mr. Guthrie whether, in the course of his researches, he came across any figures for the fuel or steam consumption/i.h.p./hour.

It is comforting to note that the early cylindrical boilers never exploded but merely "rent or opened" to relieve the excess pressure and were repaired by internal lead patches, rather in the same way as tarpaulins or collision mats are sometimes used as a temporary measure to cover up a hole in a ship's side, and one marvels at the courage of the engineers of the day who cheerfully went to sea knowing that their boilers must "necessarily give way sooner or later".

One must also admire the vision and skill of the men who planned and built this outstanding ship and it seems sad to think that she was broken up after a life of only 30 years when no doubt her hull at least was still in good condition, as these iron ships practically never wore out.

In 1946, while at Stockholm, I did a Special Survey on an iron ship built in 1879—just 20 years after the *Great Eastern*. She was then 67 years old and the main structure of the hull was in remarkably good condition for its age. She went through two more Special Surveys after that and was finally broken up in 1957 after 78 years' service—not because she was worn out, but because she had been so badly damaged in a collision on 19th April, 1957, that the cost of repairs would have been prohibitive.

This ship was built by Doxfords, as also were her engines which were of compound expansion type and still running quite well when I saw them in 1946 but were replaced at the next S.S. in 1951.

It is interesting to note that, in the 20 years which elapsed between her construction and that of the *Great Eastern*, boiler pressures had only risen to 75 lb./in.^{2}, in spite of the arrival of the cylindrical boiler and the multiple expansion engine.

Her original boilers were replaced in 1924 when they were 45 years old, the new ones having been built in 1892, so that they in turn, gave 65 years' service before she was broken up.

I should like to tell a story about the survey of old ships which, while it has nothing to do with this paper has, I think, a certain interest.

It concerns a Special Survey of a steel ship about 40 years old. I had drilled and hammered my way round the hull for two or three weeks. I had drilled the shell-plating, the decks and 'tween-decks, the bulkheads and bunkers. I had lain on my back in the dry tank under the boilers admiring the delicate tracery of what had once been floors and intercostals and wondered what was holding the boilers up. The repairs cost something of the order of £20,000 and, when I went down the gangway for the last time, I had the comfortable feeling of a job well done and confident that she would safely run another four years.

It so happened that the Yard Manager was coming up the gangway and I turned round to have a word with him when, chancing to glance at the ship, I was dismayed to see a light shining through one of the shell-plates in the bridge 'tween deck space. I drew the Yard Manager's attention to this unusual sight and asked him if he had so little regard for his reputation that he would let a ship leave his yard with a hole in her shell-plating. He glared at the offending light, said "———!!

———!!", which may be freely translated "———!! ———!!" and sent for the foreman to put on a doubler.

The moral of this story is not, as you might think, that "Pride goeth before a fall" but that there can be occasions in the career of a Surveyor when a little luck is worth more than experience and academic knowledge. The converse is also true, of course.

Mr. J. BURTON DAVIES

I had not intended to take part in the discussion on a paper concerned with machinery, but in his introduction Mr. Guthrie mentions sufficient of the general construction of the vessel to give me my excuse to take part. I have always been interested in this ship since there is no doubt whatever that she was a most epoch-making design.

I would take issue with the Author in the relative credit he has assigned to I. K. Brunel and Scott Russell. I think the recent biography of Brunel by L. T. C. Rolt gives a lot of information obtained from contemporary documents showing that the whole concept of the design was due to Brunel, although of course she was actually built by Scott Russell. It might even be truer to say that the construction was commenced by Scott Russell rather than that she was built by him, since, owing to financial difficulties, Scott Russell withdrew from the contract before the vessel was completed.

One minor point in the first paragraph of the particulars of the vessel—the *Great Eastern* was the only *merchant* vessel having five funnels; there was a Russian five-funnel destroyer of First World War vintage.

The boiler pressure of 25 p.s.i. certainly appears low when it is realised that in the conditions to be fulfilled by locomotives taking part in the trials for the Liverpool and Manchester Railway in 1829 one was that the pressure was not to exceed 50 p.s.i. It must, however, be admitted that locomotive boilers exhibited a startling tendency to blow up at various parts of the country, so perhaps the figure of 25 p.s.i. was only slightly on the safe side.

Mr. G. DIXON

The Author is to be thanked for his most excellent paper giving a very comprehensive description of the machinery of this famous ship. From the various illustrations it is obvious that the hull design is also of historic interest and a complementary paper by one of our Ship colleagues would be welcomed and complete the Staff Association's records of this remarkable ship. The repair to the 83 ft. hole in the starboard bilge at New York using a cofferdam merits inclusion. The rock which caused this damage was uncharted and only subsequently named the Great Eastern Rock. Thus the *Great Eastern* made a contribution to American geography.

Mr. Guthrie has given a brief history of this great ship and those interested in the full story of her adventures should read James Duggan's book "The Great Iron Ship".

The ship, as her name implies, was originally intended for the far eastern service and opinions have been expressed that had this been adhered to the story of failure might have been turned to one of great and well deserved success.

This monstrous iron ship caught the popular interest in the 1850's and had an almost awe-inspiring effect. After 100 years have passed by even the 200,000 ton ship, recently referred to in the opening address of the Welding and Shipbuilding Symposium has not the same effect on the popular imagination unless, perhaps, it is to be a nuclear powered hover-craft capable of skimming over the waters at some 100 knots!

The *Great Eastern* was the forerunner of many large liners with steam reciprocating machinery as was the famous *Mauretania* for the subsequent steam turbine driven liners. This age seems to lack the bold spirit of adventure which I. K. Brunel showed in the 1850's and Andrew Laing and C. A. Parsons half a century later, or is it that we have reached a point in marine engineering beyond which no revolutionary avenues are left to explore? The progress made in so many fields of engineering over the past 50 years seems to indicate that this is not the case and if the Queen 3 is built, either with government aid or by private venture, it is hoped she will be as outstanding as was the *Great Eastern* and pioneer a new era in ship propulsion.

Mr. C. DEARDEN

The Author is to be complimented on his effort in collecting and presenting particulars of the machinery installation of this great ship. She was undoubtedly the most impressive and technically advanced vessel of her day, which reflects greatly to the credit of her designers and builders.

Considering the marine machinery manufacturing and transport facilities of that time, the construction and assembly of the huge components of the engine must have been a task of the first magnitude and its successful accomplishment speaks for itself. It is interesting to reflect that probably much of the material for this great ship was carried by horse and cart through Fenchurch Street on its way to the Millwall yard where the vessel was built.

The description given on page 1 column 2 about passenger ships in the year 1860 is artistically apt, and not without its parallel 100 years later when one thinks of the giant aircraft which cross the Atlantic daily—"Pillars of smoke by day and pillars of fire by night".

The design of the boilers is interesting. I think we could use the American phrase "Ob-round" with the meaning in this case, and I shudder to think of the "Stand-out" calculations that the designer carried out. I suppose we could call this type of boiler early Gothic.

I would view with marked reservation the

approval of such a boiler plan if it were ever presented on the fourth floor and I would carefully mark in red ink "approved—subject to the boiler being constructed, installed and hydraulically tested all to the Surveyors satisfaction".

The remarks attributed to Jacob Perkins on page 19 at the end of column one and at the beginning of column two respecting Mr. Oliver Evans's plan "which never explodes", this phrase has a sublime grandeur whose audacity is breath-taking, certainly Englishmen in the age of Imperial Greatness suffered from no reservations in their remarks. Albeit in their defence it must be said that to "act effectively from incomplete knowledge is the art of living" and this they did well—their devices often did work! I might add, implicit in Oliver Evans's design is the concept of fail-safe so freely used in modern aircraft design.

The oscillating Vee type engines that drove the paddles have left a strange verbal legacy. These engines were the common type employed at sea, but were superseded following the introduction of the propeller. However, the official forms of the M.O.T. still refer to the modern piston engine as the inverted vertical type. We may assume that the official mind still regards crankshafts superimposed above the cylinders as the right way up.

The power/weight ratio of these engines makes an interesting comparison between then and now, for an engine weight of 836 tons which approximates to the present-day figure of a large-bore diesel developing say 25,000 b.h.p. the ratio is eight times the power produced by the *Great Eastern*.

In conclusion, let me thank the Author for an historically informative paper.

AUTHOR'S REPLY

To Mr. H. B. Siggers

Mr. Siggers highlights what must be the most intriguing feature of the *Great Eastern* to any marine engineer: the sheer size of her paddle engines, and the *Olympic's* wing engine used in the illustration was merely inserted as a yardstick to give scale to the picture. The *Olympic's* engines were indeed enormous, and I can well believe the story of the apprentice saying it took three of them to lift one of the bottom end nuts. This, of course, is not a reliable criterion, as I have yet to meet an apprentice who can lift anything heavier than a pay-packet. However, to give an idea of the enormous size of the *Olympic's* machinery, the following particulars might be of interest:—

- Piston rod diameter 14 in.
- Connecting rod diameter 13–15 in. at 14 ft. centres.
- Bottom end bolts $8\frac{1}{2}$ in. diameter over thread.
- Crankshaft diameter $27\frac{3}{4}$ in.
- Crankweb thickness $20\frac{1}{2}$ in.

This engine, of course, developed some 17,000 i.h.p. or five times that of the *Great Eastern* paddle engine; the components are therefore correspondingly heavier. In Scott Russell's model of his paddle engine in the Science Museum at South Kensington, the bottom ends appear to be of very light construction, and should offer no serious problem to the engineer of those days. The heaviest items were the four oscillating cylinders, each mould for which took 34 tons of iron, and the "intermediate shaft", i.e. that part of the crankshaft comprising the air pump crank and one crank with crankpin at each end, which weighed 40 tons and was only completed after the third attempt at a cost of £100 per ton. As regards overhaul, it is doubtful if either the cylinders or the shaft were ever lifted, and repairs would be confined to the comparatively light bearings.

In the case of the reciprocating screw engine the crankshaft was forged from the solid, and was fitted with a balance weight, not on the crankwebs, but on the shaft between the two cranks.

In addition to the 12,000 tons of bunkers, the *Great Eastern* had space for some 6,000–7,000 tons of cargo, but she was in the main a passenger vessel and it was in the passenger trade that she was expected to pay her way. In the old days of vacuum engines with steam at 5 lb./sq. in., say in the 1840's, the coal consumption was as high as $4\frac{1}{2}$ lb. per hour per i.h.p. With the increase in steam pressure, although still single expansion, engines in the '60's consumed about $3\frac{1}{2}$ lb. per hour, dropping to about $2\frac{1}{2}$ lb. in the '70's with the advent of compounding. With another fixed point of $1\frac{1}{2}$ lb. per hour for a triple expansion engine at 180 lb., we can derive a fair curve through these points which will give a coal consumption of $3\frac{3}{4}$ lb./i.h.p./hour for an installation using steam at 25 lb. on single expansion.

Mr. Siggers' remarks about the hazards of boiler explosions lead one to wonder what sort of supermen were the engineers of those days! As an apprentice in Glasgow, I spent several spells in overhauling two Clyde paddle steamers with oscillating engines built in the '60's of last century. The engines were controlled from the deck level, with lots of polished brass and mahogany around, but down below in the engine room there were no guard rails round the swinging cylinders or the valve gear, the flooring was of loose boards, part missing, and the only light available was a smoking duck-lamp and one had to hold a lighted match very close to this to see if it was still lit.

To Mr. J. Burton Davies

Mr. Burton Davies queries the justice in the statement at the bottom of page one in the paper, as to who built the *Great Eastern*: Brunel or Scott Russell? I confess my views were coloured by those of Scott Russell's biographers, but I am also supported by his chief draughtsman, C. H. Jordan, who later became a principal surveyor to this Society, and who gives the very strong impression that while Brunel supplied the general idea of the *Leviathan* (her original name) it was Scott Russell's firm who developed this idea and evolved the plans and all the calculations. Again,

Brunel was a railway engineer while Scott Russell was a professional shipbuilder, although it is interesting to note that both men were civil engineers: the day of the naval architect had not yet arrived.

The point about the *Great Eastern* being the only vessel ever to have five funnels is noted: I have a vague recollection of an old N. European destroyer having five funnels, euphemistically known as the "Packet of Woodbines".

Boiler pressures for locomotives were very much higher than those obtaining on steamers. Stephenson's No. 1 locomotive built in 1825 ran at 50 lb. The "North Star", also by Stephenson, in 1837, used 50–60 lb. The famous locomotive "Coppernob" built in 1846 by Bury, Curtis and Kennedy, ran at 110 lb. But it must be borne in mind that a locomotive boiler and engine are very much smaller than their marine counterparts and are correspondingly stronger for the same thickness of material. This does not fully explain, however, why pressures were so very much lower at sea than on land, for the tendency for the boilers to blow up was not confined to locomotives. It is possible, however, that marine boilers, using salt feed, scaled up and corroded so rapidly as to render them dangerous after only a few years in service.

The naval practice in the early days of steam was to scrap copper boilers after nine years' service, and iron boilers after three years'. This was simply because copper plates were rolled larger than iron ones, and it was therefore possible to build a combustion chamber of copper without a riveted seam over the hottest part of the furnace. On page 16 of the paper it is stated that the boilers (being parallelepipeds), were stayed in all directions, and reference to Figs. 19 and 20 will show how closely spaced were these stays, also how difficult of access for cleaning and inspection. Marine boilers of the day were usually supported on a cement bed over the keelson, and one Victorian worthy (obviously a shipowner), naïvely writes as follows: ". . . the boilers are imbedded in a layer of cement on the bottom of the ship and can thus maintain a full head of steam even when the boiler bottom has completely wasted away . . ." The marine engineer in the early nineteenth century was not only heroic, he was clearly expendable, and this may explain the fatalistic attitude of the period that marine boilers never wore out, they just blew up.

To Mr. G. Dixon

Mr. Dixon makes a plea for a complementary paper to be written by one of our ship colleagues on the hull of the *Great Eastern,* and I think this would be a very good idea: the vessel had a number of distinctive features such as cellular double bottom, sides and upper deck, longitudinal framing, double longitudinal bulkheads, nine anchors with their hawse pipes and capstans, etc., which ante-dated the general recognition of most of these by some 50 years. To anyone contemplating the task I can heartily recommend Scott Russell's own book: "The Modern System of Naval Architecture, 1865" wherein he describes his own ship in great detail.

The *Great Eastern,* of course, contained within her the seeds of her own destruction, as her very size doomed her to failure in an age of small ships and harbours. She was built by speculators and run by speculators, none of whom had any experience of ship management. She was designed for the Far Eastern trade to break the monopoly held by the East India Company, but was never engaged on it. She was built to carry over 1,000 passengers in the days when hardly that number of people crossed the Atlantic in any one year. Again, she might eventually have paid her way on the Western Ocean run had not the Civil War broken out in the U.S.A., and deprived her of the cotton and N. American passenger trades simultaneously.

It is possible that had the *Great Eastern* been built and run by an established shipping company with a supporting fleet she might have been successful, but then an established company would have known better than to build a ship of this size.

To Mr. C. Dearden

Mr. Dearden remarks on the difficulty of handling and transporting the monstrous components of these engines from their place of build to the slipway. Another problem, a technical one this time, was that of forging the two shafts. Fortunately for the engine builders, Nasmyth had invented his steam hammer shortly before, but even so, the paddle engine crankshaft, weighing some 40 tons, was only completed after the third attempt at the then appalling cost of £100 per ton. In the construction of previous steam engines, all heavy forging had to be carried out under the trip hammer: a heavy steel block fitted to the end of a wooden rocking beam, the other end of which was depressed and released by a rotating star wheel driven by a donkey.

The passenger steamer in the 1860's was invariably a paddle vessel and conversely, the paddle vessel was inevitably a passenger steamer: the small working range of paddle immersion could not be equated with the large difference in draught between light and loaded conditions in a cargo boat. Again, paddle steamers were not the easiest ships to navigate across the Western Ocean, as the constant rolling, with the one paddle or the other constantly out of the water, made steering a difficult and arduous task. Fortunately the masters who commanded the early steamers were all lusty, mahogany-faced sailing ship men, with scant regard for comfort, and who thought nothing of battening the passengers below hatches in dirty weather, or even clapping them in irons on very little provocation. How well those tough old sea dogs compare with their present-day counterparts; these mincing, lisping personalities beloved of T.V. interviewers!

With regard to the M.O.T. definition of the modern steam reciprocating engine as the inverted vertical type, this may stem from the days of the

atmospheric engine with an open-ended cylinder and piston working the rocking beam through a chain. Here the cylinder had to be placed below the running gear in order to work at all, and to allow for a water seal on top of the piston.

In the 1920's there used to be a pumping station not very far from Glasgow with such an engine still in use, and the piston used to rise and sink with ponderous serenity, carrying with it a top dressing of old boots and broken bottles.

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THE SURVEY AND TESTING OF MARINE ELECTRICAL EQUIPMENT

by

W. MORRIS

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

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The Survey and Testing of Marine Electrical Equipment

By W. Morris

INTRODUCTION

The modern shipboard electrical installation is a far cry from those of the immediate post-war years as instanced by the tremendous increase in installed kW-generating capacity and the growing use of alternating current. The desire to take advantage of the simple squirrel cage motor, particularly for deck machinery auxiliaries, has resulted in many self-regulating alternators with a very quick voltage recovery response being produced.

With the increase in the electrical equipment fitted on board has also come the drive to reduce the weight, size and cost, and synthetic materials having the necessary physical and electrical properties are now used for a variety of purposes. Some of these materials, however, bring problems of their own, which only show up under service conditions.

The reconstituted Electrical Rules recognise these developments, and the emphasis on British Standards and practices, which formed the backbone of the old Rules, has been dropped in favour of an international approach. Copies of the various standards of the International Electrotechnical Commission referred to in the Rules should be kept in every Surveyor's Office, as these will be required from time to time.

Overseas many of these installations are installed under the survey of the Engineer Surveyor, and it is hoped that the following remarks will be of assistance to him.

The recently issued Instructions to Surveyors (Electrical) give guidance on the interpretation of the Rules for Electrical Equipment, and the scope of this paper is, therefore, intended to cover the practical aspects of the Surveyor's duties and also the problems, which occur from day to day. It is proposed to deal with these as follows:—

- (I) Factory Work
  - (*a*) Rotating Machines
    - (i) During Construction
    - (ii) Testing
  - (*b*) Switchgear
  - (*c*) Transformers
  - (*d*) Cables
- (II) New Construction Surveys
  - (*a*) During Installation
  - (*b*) Testing
- (III) Periodic Surveys

(I) FACTORY WORK

When called in for these surveys, it is important to establish that any plans such as those required for propulsion equipment, refrigerated cargo fan motors, welded shafts and switchboards have been approved.

Where a certificate is requested for equipment without any indication that it is for marine use and where its future service is unknown and no standards or specification are quoted in the order, probably only "Lloyd's Certificate required", then it should be treated as essential to classification and the Society's Rules applied.

If, however, its service is known to be a non-essential part of a marine installation such as air conditioning, cargo winches or domestic motors, the national standards or preferably the appropriate I.E.C. recommendations as allowed in paragraph M 101 can be applied.

Should, however, no indication be given as to the future purpose of the equipment and only special standards and/or specifications are quoted in the order, then the item should be checked for compliance with these order requirements, and in all such cases a Report 10 should be issued, worded in accordance with the facts and the order references quoted for identification purposes. Incidentally many such items, tested in Germany, are for shipment abroad and may be largely used for land installations.

(a) Rotating Machines

(i) During Construction

As in some countries the motors are rated in kW, it would pay the Surveyor to point out the h.p. limit given in paragraph M 427 to any new manufacturer.

The requirement of inspection during manufacture can in the Author's opinion be met if the Surveyor establishes that the normal workshop practice of the firm and also the workmanship employed and materials used are good and sound. To this end regular visits are paid and attention given to the following:—

Core Assemblies and Frames: The punching of the laminations are checked from time to time to make sure that the insulation facing is not damaged during this operation, as this could result in hot spots in the assembled core, later when the machine is in service. Insulating paper or varnish are generally used as the lamination facing, normally being applied to one side only. The cores are checked to ensure that the laminations have been accurately punched, and the slots are free from burrs. The ventilating duct spacers fitted between core packets can also be damaged when these cores are fitted, as this is generally done under hydraulic pressure, and these are also inspected to see if they are buckled or distorted.

With A.C. machines, the stator frames are often shrunk over the finished core, and the shrink fits used should be especially checked to ensure that there is no danger of the core moving in service. Many stator housings and D.C. magnet frames to-day are of welded steel construction, and with

D.C. machines the welds should be examined for porosity or slag inclusions, as these can affect the distribution of the magnetic flux in the frame.

SHAFTS: Shaft materials are checked to prove compliance with the requirements of Chapter P, and where the construction employed for attaching the core involves welding spiders or armature arms to the shaft, the welding should be at least $\frac{3}{4}$–1 in. away from a change of section. The plans should, however, be approved as required by paragraph M 102. The assembly should be suitably treated before and after welding and crack detection tests, carried out on the welds, to check the workmanship and welding methods employed.

The methods employed for fitting the cores to the shafts are also investigated as, if these are not properly secured, they could work loose in service.

COMMUTATORS: The commutator assembly of D.C. machines is examined, particularly the formed mica insulation used for the vee grooves and the mica pieces fitted between the copper segments. For very long commutators and also for those in high speed machines, it is sometimes necessary to shrink a ring in the middle of the commutator, so as to prevent any movement of the segments, and this operation and also the mica insulation fitted underneath should be checked.

Commutators are also manufactured to-day using a pressure moulded insulation and having the construction shown in Fig. 1. Cases have been known, however, of the segments moving due to centrifugal force and also the inner securing rings shorting out some of the segments.

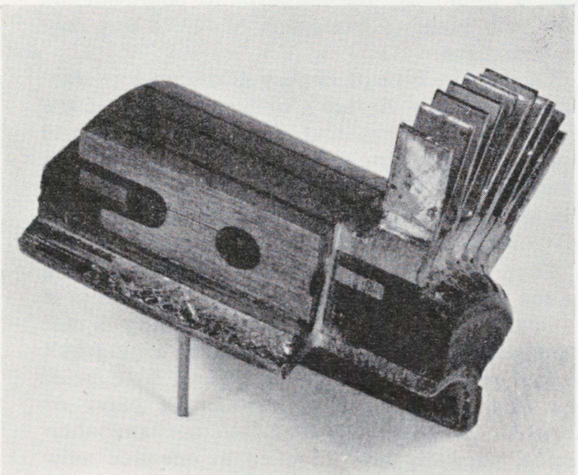


FIG. 1

Commutator having pressure moulded insulation

SQUIRREL CAGE ROTORS: The methods adopted for securing the rotor conductors in squirrel cage motors are checked from time to time, particularly where these motors are used for deck machinery drives. Brazing is sometimes used in securing the bars in position, and it is important that the coefficients of expansion of the various metals should be practically the same, otherwise there is a danger of the brazing fracturing in service. The attachment of the short-circuiting rings to the conductor bars is also examined to see that sufficient contact surface has been provided.

WINDINGS AND INSULATION: The three most important insulations in a machine are the slot insulation, the conductor insulation and the impregnating materials, and when Class A or E materials are used, the windings are checked to make sure that they are further insulated from the frames or cores with mica or similar materials (e.g., glass or bakelised asbestos) as required by the Rules for essential machines.

The insulating materials employed are selected to meet the estimated operating temperatures expected and not infrequently a machine may have different classes of insulation provided for the various windings. Thus the armature may be Class B, while the field windings are Class A. The various classes of insulation are given in the Instructions to Surveyors, and it is useful to note that the differences between Classes B, F and H are largely in the impregnating materials used, e.g., modified silicone resins being used for Class F and a pure silicone resin for Class H. These two latter insulation classes may only be used, however, with special approval.

The slots are generally fitted with liners, which serve two purposes, one being to prevent mechanical damage and the second to provide additional insulation between the windings and the core. These liners are often made in three layers, the outer layers comprising press-pan or elephantide, with a middle layer of mica or micanite. Other materials commonly used for liners to-day are asbestos impregnated with resin or a sandwich of thick glass tape insulation, micanite and polyester foil. This latter foil is also known under the trade names of "Hostophan", "Melinex" and "Mylar".

It is good practice for the liners to project from the slots and the windings extra taped in these positions, as it is here that breakdowns often occur. A stator during construction is shown in Fig. 2.

The overhung stator windings of large A.C. machines are sometimes supported by means of an insulated metallic ring. This ring should not be made of magnetic material, otherwise it would probably overheat with resultant damage to the attached windings.

There is also a growing practice to-day to competely seal the stator end windings by encasing them in an epoxy resin. The resin used should be resistant to moisture and oil attack, have satisfactory bonding properties to the materials to which it comes in contact, e.g., cables, frame and lamination, be compatible with other insulation materials used, be mechanically strong and should not crack when setting or on subsequent temperature cycling when in service.

SLIP RINGS: Slip ring connections are given attention to see that the leads are sufficiently rigid or mechanically supported and also adequately insulated, particularly where they pass through a

slip ring of opposite polarity and where they emerge from a hollow shaft.

BANDING WIRES AND SLOT WEDGES: Banding wires should be applied under tension and secured together at frequent intervals with a soldered metal tape. These are not only used to secure the end windings of D.C. armatures and A.C. wound rotors, but are quite frequently fitted instead of slot wedges (on machines where the centrifugal forces acting on the rotating windings are very low).

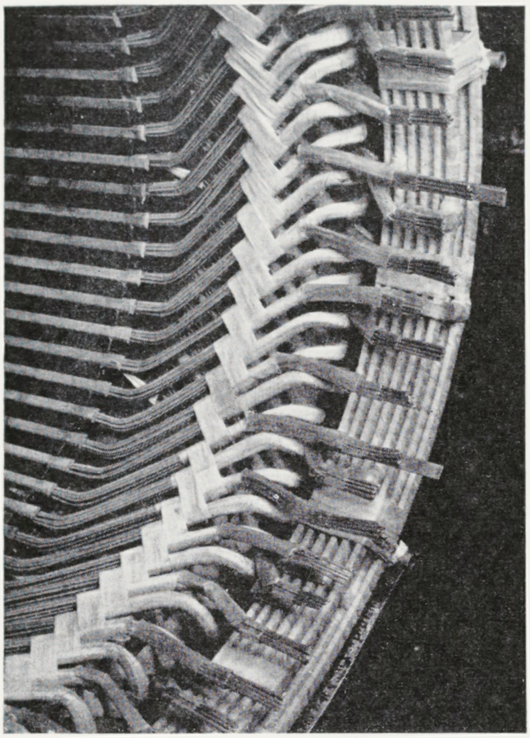


FIG. 2

A.C. Stator Winding during construction (Hans Still A.G.)

The slot wedges should be a smooth drive fit, and wood, phenolic resin materials (e.g. "bakelite") and impregnated fibrous materials are used. They should be examined before and after the impregnation process.

IMPREGNATION: To ensure that the windings are protected against ingress of moisture, oil and dirt and from fungus attack, it is necessary to impregnate them with an insulating varnish. There are several methods of doing this, and they include hosing, ladling, flooding, dipping and vacuum impregnation. The windings are generally preheated before treatment, then allowed to drain off before being transferred to a baking oven for curing. The duration of the process depends largely on the size of the windings and in some cases is repeated two or three times. The methods most often employed are hot dipping and vacuum impregnation.

It is also the practice of some manufacturers to brush or spray all windings and connections of finished A.C. machines with an insulating varnish or paint after the impregnation process, as they consider that this seals the machine electrically and helps to maintain the initial insulation resistance values.

A hard glossy finish is always desirable as any dust accumulation can be easily blown off.

SELF-REGULATING ALTERNATORS: The regulating equipment is quite often built on to or even into the machine, and a typical arrangement is shown in Fig. 3. The transformers, coils and rectifiers are examined to see that they are securely supported and adequately ventilated. It should not be forgotten that in some cases the hot air from the machine passes over or near this equipment, and it should, therefore, be rated to deal with this high ambient temperature.

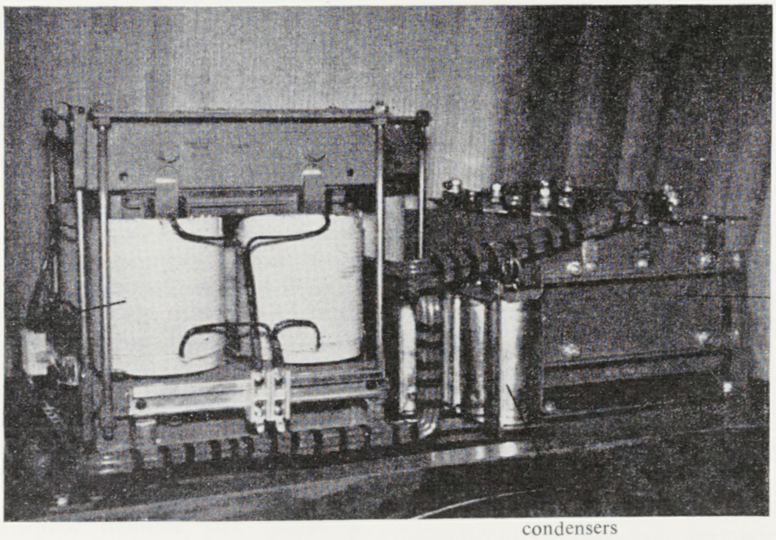


FIG. 3

Self-Regulating Device fitted on A.C. generator (Hansa Motoren)

A brief description of a self-regulating alternator will probably be of interest, and the diagram of connections of a Siemens machine is shown in Fig. 4.

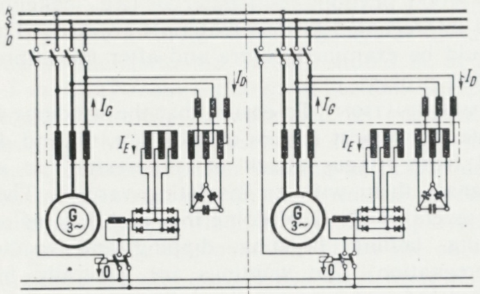


FIG. 4

Siemens Alternators connected in parallel

The excitation arrangements may be compared with those of a D.C. compound wound generator, because the excitation comprises a load independent ("shunt") and load dependent ("series") component. The field winding is excited from a rectifier, which is in turn fed from the secondary winding of a combining transformer having two primary windings. One is supplied from the alternator terminals via a reactor and is the no-load component, while the second carries a current transformed from the alternator output current and is the load component. These two components are added vectorially in the secondary winding, and the resulting current is rectified and fed to the exciter windings. Fig. 4 also shows the arrangements adopted for parallel operation, and the comparison with D.C. compound wound generators may be taken one step further, as Siemens provide an "equaliser" connection, to which the field windings are connected through double pole contactors.

RADIO SUPPRESSORS: Finally, as most D.C. machines are to-day fitted with radio suppressors, all generators and steering gear motors are checked to see that these suppressors are either provided with built-in fuses or are protected with separate fuses.

(ii) TESTING

Where duplicate machines are to be tested, it is not always necessary that all should be given full tests. By applying the following formula and comparing the result with the figure in M 428, it can be established whether abbreviated tests may be carried out:

$$\text{kW/kVA/h.p. per 1,000 r.p.m.} = \frac{\text{rated kW/kVA/h.p.} \times 1{,}000}{\text{rated r.p.m.}}$$

This stipulation is really on the machine size, as the physical dimensions of a machine vary inversely as the speed.

The following are the full tests normally carried out:—

(1) Temperature rise and commutation test.

(2) Momentary overload (or overcurrent) tests.

(3) (a) Voltage regulation tests (generators),

(b) Characteristic tests (motors).

(4) High voltage tests.

(5) Insulation resistance tests.

(2), (4) and (5) are straightforward and do not require comment.

Sudden short circuit tests are usually carried out by manufacturers on all new types of A.C. generators. These tests are carried out to check the mechanical stability of the machine, particularly the end windings and also to derive the machine constants which determine the values of the instantaneous short circuit current and the rate at which this current decays. In addition, the transient voltage response of self-regulating or A.V.R. controlled alternators at various suddenly applied loads (i.e., switching on of large squirrel cage motors) is also tested. These tests generally form part of the manufacturers' investigation programme, and as they are not normally witnessed by the Surveyors, they are considered to be outside the scope of this paper and are consequently not included.

(1) TEMPERATURE TESTS

The duration of this test should be long enough to establish that the temperature rises are sensibly constant, and it is the Author's practice to accept as proof that this is so, when they do not increase by more than 1°C./hour. The normal duration is about four hours, although cases do occur when a longer period is necessary. Some manufacturers, however, run the machine on overload for a time and then reduce the load to the full load rating. In this way the test duration can be shortened.

It is, however, necessary to know what the temperatures of the machine are when it is running, and as a guide the inlet and outlet air temperatures, the frame temperatures and also the temperatures of the stationary windings are measured by means of thermometers. When these temperatures are constant, the machine may be shut down and measurements taken. It should not be forgotten that the temperatures will rise for the first 2–3 minutes after shut-down, thereafter dropping in value. Where forced ventilation or separate water cooling is employed, these should be shut off at the same time as the machine is shut down.

The temperatures are normally measured with thermometers or by the increase in resistance method. With this latter a figure of 2·5° C. per cent increase in resistance is helpful in roughly establishing the temperature rises.

Where thermometers are used, their bulbs should be covered with a pad of felt or cotton or similar non-conducting materials to prevent loss of heat by radiation or convection. Putty or plasticine can also be used. Another method employed with commutators is to wrap the thermometer bulb with a metal foil, such as silver paper found in cigarette packets, and arranging this with a projection, which can be placed under a brush. The ambient temperature should be the average of several measurements made at distances of between 3 ft. to 6 ft. from the machine.

Small machines can generally be loaded without difficulty, as a suitable motor or generator with a load circuit of resistances or water tank can usually be provided.

D.C. MACHINES: In the case of large machines, it is more economical as well as more convenient to employ a second machine for absorbing the output of the first. If the machine to be tested is a generator, then the second machine operates as a motor, driving the first machine, or vice versa. This is known as the "back to back" combination, in which the generator provides the electrical power for the motor. In this way large pairs of machines can be tested under full load conditions with only a very small supply being drawn from the mains to make up the losses.

It should, however, be remembered that the field currents are not at their normal values during this test and that further, if the machines being tested are generators, then the machine operating as a motor will be doing so under an overload of approximately 20 per cent.

A.C. MACHINES: The above method is also applied to synchronous machines (i.e., having D.C. excitation). Another method is to short circuit the main output terminals and with reduced excitation cause full load current to be passed through the machine. This is very often the only practical way to carry out temperature tests on single large machines in small works as their testing facilities do not permit any other method. It is not a true full load test, as the excitation windings are only lightly loaded and the normal iron losses produced by the rated voltage are not present.

Two further heat runs are then generally carried out, one on open circuit at rated voltage and the other at the maximum excitation current permitted by the design staff, bearing in mind the resultant generated voltage. From these three results, the temperature rises at rated load and voltage are calculated.

A further method of testing a synchronous machine is to operate it in parallel with the mains and adjust the excitation current so that the power factor of the load current is zero, or to provide an inductive load to give the same effect. By this means a full load temperature test can be carried out, although in this case the excitation current will be greater than that required for the normal full load at rated power factor.

If the temperature rise of the excitation winding is found to be above that permitted, the rise at rated excitation current can be calculated to within a degree or two by reducing the observed temperature rise in the proportion of the square of the two currents.

With large asynchronous (e.g., squirrel cage) motors, there is very little alternative to loading them through attached generators.

COMMUTATION: The commutation of the D.C. machines should be examined during the full load test, and the sparking, if any, should only be, at the most, pin-point sparking.

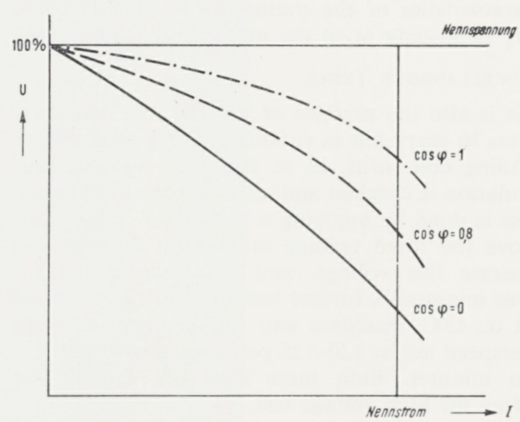
(3) (a) VOLTAGE REGULATION TESTS (GENERATORS)

Variations in voltage and frequency affect the output and torques of motors, and unless these are kept within reasonable limits, the machines could overheat with normal load.

Voltage regulation tests can only be carried out, when a driving motor (or engine) of the necessary output is available to drive the generator at full load. In the case of A.C. machines, the associated automatic voltage regulator and exciter or the self-regulating equipment is also required, together with a suitable inductive load, as these generators are designed to operate at a definite power factor, generally between 0·75 and 0·8 lagging. The inductive part of the load is generally provided by a reactor with an adjustable air gap, and this with a water tank or resistances enables the selected load to be applied. Normally this is done in steps of 20–25 per cent up to 125 per cent full load, and the generator voltage measured at each step.

In carrying out this test, it should not be forgotten that the speed of the generator set at no-load is generally about 4 per cent higher than at full load, the exact amount being dependent on the accuracy of the engine governor. If the tests are, therefore, carried out employing an electric motor as the driving medium, then the speeds should be adjusted accordingly.

This voltage characteristic should then be checked to see that the permitted tolerance of 2·5 per cent is not exceeded. Voltage curves of an alternator at various power factors are shown in Fig. 5.



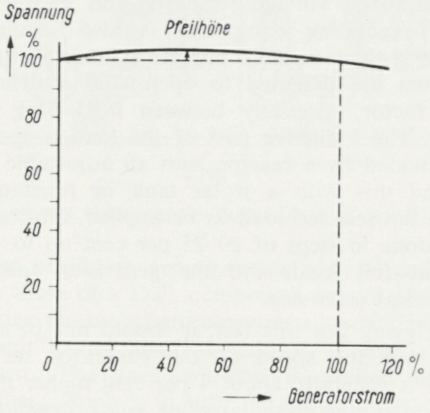
U = Voltage
Nennspannung = Rated voltage
Nennstrom = Rated current

FIG. 5

Voltage Characteristics of alternator at various power factors and at constant no-load excitation (Siemens)

Where the prime movers are available for this test, the governing of A.C. sets intended for parallel operation with one another is particularly checked to see that their permanent speed variations are equal within a tolerance of $\pm$ 0·5 per cent.

The field currents are set at 20 per cent full load to give rated voltage and are not adjusted during the test. The permitted voltage difference between these load values is 2·5 per cent. It should be noted that the "hump" in the D.C. compounding curve is also limited as indicated in paragraph M 417 and is likely to be between 3 and 5 per cent. A typical voltage curve for a compound wound D.C. generator is shown in Fig. 6.



Spannung = Voltage
Generatorstrom = Generator Current
Pfeilhöhe = Voltage Rise

FIG. 6

Voltage Characteristics of D.C. compound generator (Siemens)

(3) (b) CHARACTERISTIC TESTS (MOTORS)

Tests are carried out to establish that the characteristics of the motor are as stated. These vary depending upon the motor application.

MISCELLANEOUS TESTS

It is also the practice of the German manufacturers to carry out as routine a voltage test on the winding coil turns, as in this way the interturn insulation is checked and any weaknesses exposed. This is done by applying a voltage of 30 per cent above the rated voltage or making the machine generate this voltage, and it is maintained for three minutes. A further test also normally carried out on D.C. machines and A.C. generators, is an overspeed test at 120–125 per cent rated speeds for two minutes. Both these tests are carried out before the high voltage test.

(b) Switchgear

Whilst, apart from propulsion equipment (M1701), it is not a requirement of the Rules that switchboards and similar equipment be inspected and tested at the manufacturers' works, these surveys are often requested.

Two types of construction are employed, namely, the open type and the dead front type. With the open type, the circuit breakers, switches and fuses and other ancillary equipment are mounted on the front of an insulation panel with all "live" parts exposed. A typical example is shown in Fig. 7. Marble and slate may still be used in the construction of switchboards, provided the "live" parts are insulated therefrom with mica or similar insulating materials.

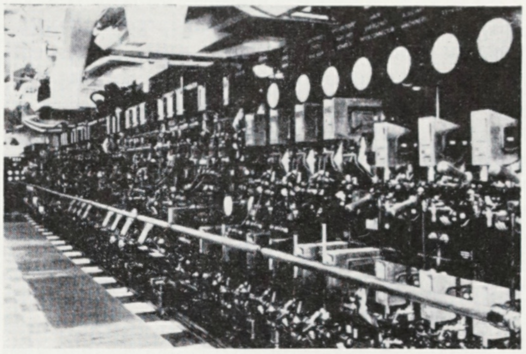


FIG. 7

British Open Type Switchboard

The dead front switchboard is usually of an enclosed sheet metal and angle iron construction, where the "live" parts are only accessible from the back or through hinged or screwed panels, fitted to the front of the board. The arrangements frequently adopted are shown in Figs. 8 and 9. This type of switchboard is specified by the Rules where the voltage between poles or to earth exceeds 150 volts A.C. or 250 volts D.C. (paragraph M 601(c)).

The following remarks apply to both types and also to similar equipment.

Firstly, fuses and circuit breakers are to be of an approved type and details of these should be shown on the plans submitted for approval.

INSULATING MATERIALS: All insulating materials employed in the construction of the switchboard and its auxiliary equipment, such as the breakers, switches, etc., to which "live" parts are attached are to be resistant to tracking, moisture, sea air and oil vapour. They do not, however, have to be an approved type as previously required, but they should be checked to see that they possess the necessary qualities. The only practical way to do this, if in doubt, is to ask for the manufacturers' test certificate. There are many types used to-day, and these may in general be split into two categories, namely:—

(1) Compressed asbestos with suitable binder.

(2) Synthetic resins with mechanical reinforcement.

Well known examples of the compressed asbestos type of insulation are "Sindanyo" and "Interohm" insulating panels, generally used in the United Kingdom for open type switchboards.

There have been many advances made in the development of synthetic resins, but it is outside the scope of this paper to go into these materials in detail. Most countries have their own standards in which tests to prove the mechanical, electrical, water absorption, heat resisting and inflammability properties are given.

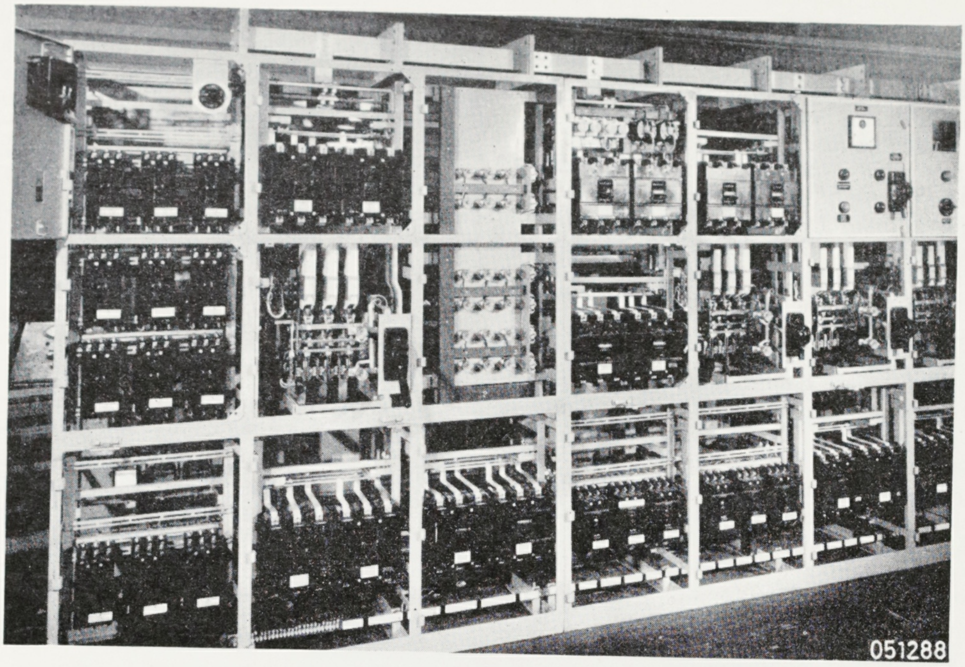


FIG. 8

A.E.G. Dead Front Switchboard (front) (with doors removed)

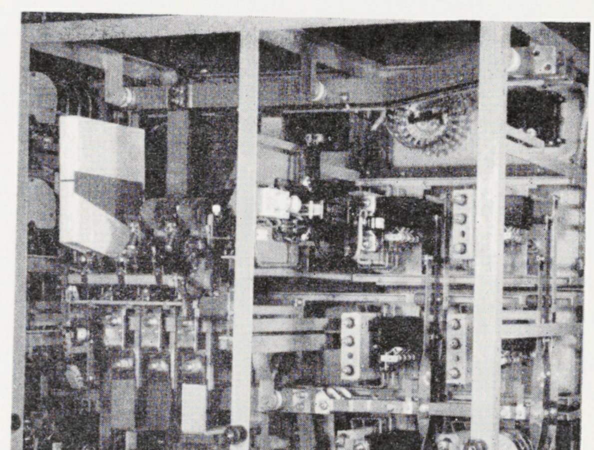


FIG. 9

A.E.G. Dead Front Switchboard (back)

Phenol - formaldehyde synthetic resins or "bakelites" and aminoplastic resins or "melamines" as they are more commonly known are representative of the second group and bonded insulation materials made from them are used in the manufacture of switches, circuit breakers and other switchgear parts. Their electrical properties are superior in many respects to those of natural resins. The "bakelites", however, have a tendency to track, and for this reason melamine insulating materials are preferred by many Continental manufacturers.

This anti-tracking property of insulating materials is important, in the Author's opinion, particularly where the clearance and creepage distances are small. Tracking occurs when the leakage current between two adjacent parts of opposite polarity causes the surface to carbonise. This can be cumulative and result in a complete failure. A recommended test for determining the anti-tracking properties is given in the I.E.C. Publication No. 112 (1959).

CONSTRUCTION: The switchboard should be carefully examined to make sure that all connections are tight and secured against working loose due to vibration. There are many ways of doing this, such as locking nuts, patent lock washers, etc., or, with small screwed connections, by painting them with shellac. All parts should be accessible for inspection and maintenance.

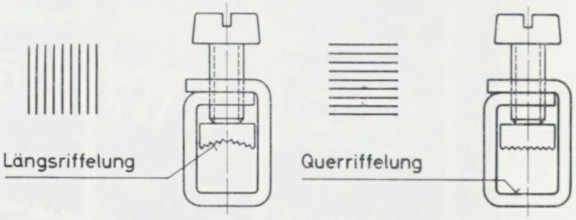
Any resistances, such as generator field resistances or those required for battery charging circuits, which are fitted as an integral part of the board, should be so installed that there is no danger of the heat emanating from them causing trouble to adjacent cables, fuses or similar items. These remarks also apply to transformers built into the switchboard, as this is frequently done to-day, e.g., lighting transformers or those required for self-regulating alternators.

The generators and outgoing feeder circuit connections should be arranged so that the cabling on board can be easily accomplished and the ideal, of course, is to bring them either to the top or bottom of the switchboard, thus eliminating site wiring to the individual switches or circuit breakers.

WIRING AND CONNECTIONS: P.V.C. and butyl-insulated cables are often preferred by manufacturers to-day to rubber insulated cables. The methods of running and securing these cables are many and various. Some are run exposed and clipped, others are harnessed to back straps or rods while in other cases they are placed in channels, made of insulating materials, being held in position by a suitable cover. This arrangement has certain advantages inasmuch as the cables can be easily removed, if necessary. Whichever method is employed, however, the cables should be secured against vibration. It should also be remembered that for propulsion equipment a minimum cross-sectional area of 0·0045 sq. in. (2·5 mm.^{2}) is required for essential connections.

Compression type cable shoes are preferred by the Author to the soldered cable shoes, Ross Courtenay or similar connectors. It is, however, important to see that the work tools used are of the correct design, such that when a start has been made in "making off", the tool cannot be released until the full pressure has been applied.

Patent type cable terminals, which clip on to a supporting rail, are frequently used, and the principle of operation is shown in Fig. 10. The clamping screw is locked by means of the spring reaction of the yoke. Where these are used for V.C. cables, a crimped or similar metal sleeve should be provided for the conductor or the ends soldered so that the cable end can be made watertight.



Längsriffelung } Serrated Surfaces
Querriffelung }

FIG. 10

Klippon Terminals

SWITCHES AND FUSES: All fuses should be located as near to the source of supply as possible and where rotary switches are used, the Author considers that the fuses should be fitted between the busbars and the switch, as with this type of switch it is impossible to check its condition by visual examination. A fault, therefore, in one of these switches could become a busbar fault.

Such switches and also combination fuse-switch units are often fitted where space is at a premium. With these latter, the fuses are attached to the knife blades. The securing arrangement of the stationary contacts of these and other knife switches should be such that they cannot rotate on their own fitting bolts, and this can be prevented by fitting off-centre dowels.

CIRCUIT BREAKERS: Where miniature circuit breakers are provided for lighting and small power circuits, "back-up" short circuit fuse protection is sometimes required due to their low breaking capacities, and they are generally supplied in groups, through fuses having a maximum capacity of between 60 to 100 amperes. This fuse rating is specified by the circuit breaker manufacturer. Where the generating capacity of the installation is large, such "back-up" protection is sometimes required for outgoing circuit breakers and motor contactors, when these are fitted as part of the switchboard. All feeder interrupting devices at the main board should have equal interrupting capacities. This aspect of the installation is, however, checked by the Plans Department, when the plans are submitted for approval.

A.C. switchboards in the United Kingdom are commonly fitted with draw-out circuit breakers, but this practice is not favoured on the Continent.

Isolating switches or links between the breakers and busbars are provided, when specified, and "back-up" fuses, when fitted can also serve the same purpose. The trend to-day in any case appears to be towards the moulded case type of circuit breaker due to its compactness and relative cheapness, and a typical example is shown in Fig. 11.

All circuit breakers are to be trip-free (i.e., they trip themselves free instantaneously when closed on a fault) and also fitted with overload and short circuit trips. The overloads may be thermal or magnetically operated and for generator breakers, they must also be adjustable. Preference tripping circuits are also to be provided with adjustable overloads so that the tripping can be staggered over about 5–20 per cent overload.

Some manufacturers design their preference tripping arrangements on a time and not current basis by providing auxiliary contacts on the generator circuit breaker overloads, which operate about 10 seconds before those provided

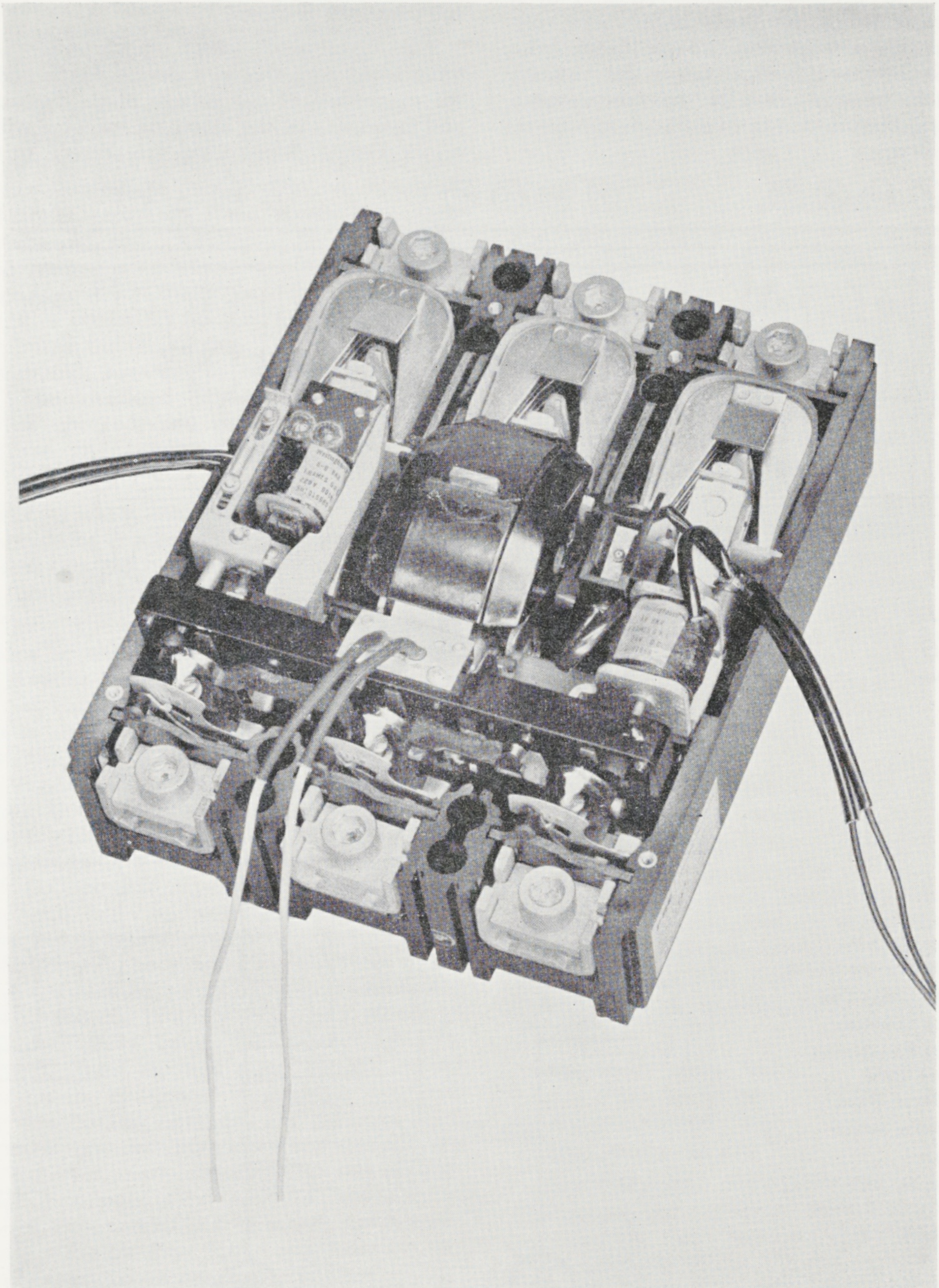


FIG. 11
Moulded Case Circuit Breaker (FPE)

for tripping the generator. This is, however, no longer permitted by the Rules.

The reverse current trips on D.C. generator breakers should also be carefully checked to make sure that they are in the pole opposite to the equaliser. If this is not so they will not operate correctly as the generator current could be shunted through the equaliser connection resulting in only part of the reverse current passing through the reverse current coil.

SYNCHRONISING ARRANGEMENTS: The normal practice is to provide a central panel with a synchroscope, lamps, frequency meters and voltmeters but the arrangements adopted vary from country to country. It is important in the Author's opinion that an alternative to the synchroscope (e.g., lamps) be provided in case this should break down in service.

In some cases, as shown in Fig. 12, each alternator panel is fitted with two synchronising lamps, a double frequency meter and a voltmeter. A further method employed is to switch the generator to the busbars via a contactor and a choke without first synchronising the voltage phases of the incoming machine to the busbars, but this is generally only done with self-regulating alternators. The diagram of connections for the A.E.G. "Synchromat", which employs this circuit, is shown in Fig. 13. The circuit breaker is generally interlocked so that it cannot be closed, unless the contactor is closed.

The "Austinlite" check synchroniser is also often fitted, and this is a device which compares the magnitude of the voltage phase displacement and frequency of the incoming machine with that of the system. When these are correct, to within

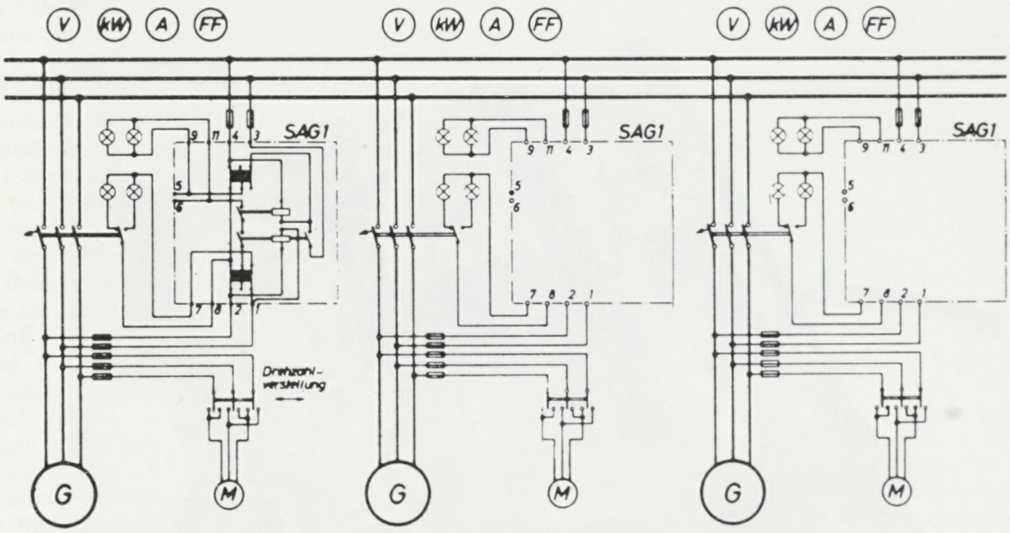
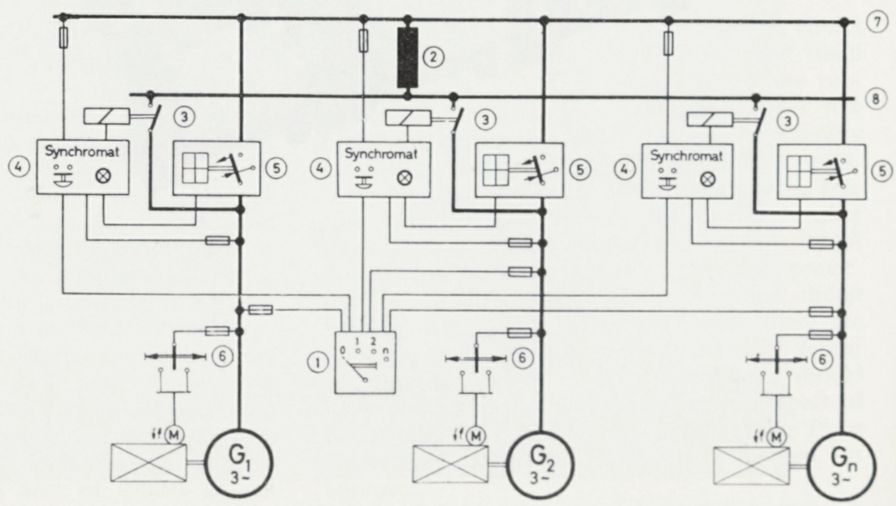


FIG. 12

Siemens Synchronising arrangements

Drehzahl verstellung = Speed Regulating Switch



1 Synchronising Selector Switch
2 Synchronising Choke Coil
3 Synchronising Contactor
4 Synchronising Push-button and Indicating Lamp
5 Alternator Circuit Breaker
6 Speed Regulating Switch
7 Main Bus Bar
8 Synchronising Bus Bar

FIG. 13

A.E.G. Synchromat

safe limits for paralleling, a pilot lamp lights up and the breaker may be closed.

EARTHING: The switchboard should be checked to see that the secondary windings of the instrument transformers are earthed and that these connections are readily and easily identifiable.

The detection of earths on the system is done by means of lamps or a voltmeter, connected to earth through a selector switch. The voltmeter is calibrated in ohms and is generally referred to as an ohmmeter.

CONTACTOR PANELS: With automatic motor contactor equipment, such as is sometimes provided for refrigeration installations, the control circuit of one drive is often interlocked with another. In such cases it is not possible to completely isolate the circuits individually without fitting switches with many auxiliary contacts, and these can complicate the system. A common control circuit, separately fused, is sometimes provided to avoid this, but should this circuit be supplied from a transformer, a second unit with a change-over switch should be fitted as a standby to guarantee continuity in case the working transformer burns out in service. Guidance on such circuits should, however, be obtained from Head Office. The overload protection arrangements should be checked, and further details on this subject are given under "Starters and Control Gear".

LABELS: Finally, the fuse label on the equipment should be checked to see that the cable size or current carrying capacity and also the capacity of the fuses are given.

TESTING: The effects of voltage and frequency variations on switchgear are quite important, and where possible these should be checked.

A.C. contactors are likely to overheat if the voltage is high or the frequency much lower than normal. On the other hand, if the voltage is low or the frequency high, the contactor may not have sufficient closing pull and will become noisy, because it will start to open each time the voltage passes through zero.

With D.C. contactors, if the voltage is low, the closing pull will be correspondingly smaller, whereas if it is high the coil might overheat.

Vibration and shock are also important to the effective operation of switchgear as a combination of these, together with voltage and frequency fluctuations could cause maloperation of the equipment. Tests are not often carried out on normal marine equipment to establish their suitability to withstand vibrations and shock as, apart from naval practice, no values are specified for these. However, most manufacturers rely on their experience obtained over the years and also the results of tests carried out to satisfy naval specifications, although these latter conditions are not normally expected on merchant vessels.

Tests should also be carried out as far as possible to prove the operation of the protective devices, and this generally means functional tests on the control or secondary circuits as very few workshops are provided with the necessary facilities to directly test overload or short circuit releases. With A.C. switchboards, the secondary circuits can be injected with the necessary current, generally about 5 amperes, and similarly a millivoltage may be applied to the secondary side of protective circuits of D.C. switchboards. Short circuit and thermal overload trips can be operated by hand to show that the releasing arrangements are in order. High voltage and insulation resistance tests should then follow.

(c) Transformers

It is sometimes necessary to witness tests on transformers, and those required are given in M 908. The construction, workmanship and materials can be checked at the same time. Liquid cooled transformers and also those having insulation materials other than Class A or B must be specially approved.

The temperature test is generally carried out by short circuiting the secondary winding and passing a current, which results in losses in the transformer equivalent to the sum of the iron (no-load) and copper (load) losses. These losses are first established by carrying out separate tests on the transformer, first on open circuit with normal voltage applied and secondly on short circuit where the normal full load current is passed.

The temperature rise is measured by the increase in resistance method.

(d) Cables

It is in the manufacture of cables that the practices of the various countries diverge most from one another. The Surveyor overseas is often requested to examine and witness type tests on cables, and those carried out by the Author are as follows:—

(1) Methods of construction.

(2) Check on dimensions.

(3) Electrical Tests:—
- (*a*) Conductor Resistance.
- (*b*) Insulation Resistance.
- (*c*) High Voltage Tests.

(4) Mechanical Characteristics:—
- (*a*) Tensile Strength before and after ageing.
- (*b*) Elongation before and after ageing.
- (*c*) Tensile set (permanent elongation).

(5) Water Absorption Test.

(6) Thermoplastic Characteristics of P.V.C. Cables:—
- (*a*) Hot Pressure Test.
- (*b*) Cold Bend Test.
- (*c*) Heat Shock Test.

(7) Fire Test to M 850.

The acceptable test results are given in the Instructions to Surveyors and the methods of carrying out these tests are given in the I.E.C. Publication No. 92.

The make-up of the cable should be particularly checked to make sure that rubber and P.V.C. compounds are not in contact, for the reasons given in the Instructions to Surveyors.

Where heat resisting or synthetic rubber insulation is used, the cable should be readily identifiable, because higher current carrying capacities are allowed. There are several ways of doing this, such as the inclusion of markers or coloured threads inside the cable or by indelibly marking the sheath. The outside covering of the cable is also sometimes painted with a special colour for this purpose.

A simple test to check the rubber insulation is to set it on fire and to smell the smoke and examine the residue. Normal rubber and also silicone insulation can generally be very easily distinguished.

(II) NEW CONSTRUCTION SURVEYS

Every Surveyor has his own method of carrying out these surveys and the following remarks indicate the approach taken by the Author.

The plans and short circuit calculations are examined immediately after approval and any amendments entered into the F.E. booklet. Details of the generator, distribution and motor cables and also make, type and rating of the circuit breakers are also added, together with the Rule rating of the cables.

The following are then taken up with the electrical drawing office:—

(i) Plan approval amendments so as to make sure that they are being dealt with.

(ii) Types of cables to be installed, methods of installing them and their location. Rule requirements for flame extending cables are pointed out.

(iii) Remote stops for oil fuel pumps, fans and cargo pumps as there are so many of these located in various parts of the ship to-day.

(iv) Location of E.S.D. oscillators. These should not be fitted if it is at all possible, in O.F. double bottom tanks or changeable tanks. Sometimes this cannot be avoided and then special arrangements should be adopted.

(v) Equipment in dangerous spaces.

(vi) Manufacturers' test certificates for all essential generators, motors and transformers are requested. This is particularly necessary overseas, due to the various national standards and also because of the many other Classification Societies involved.

(a) During Installation

CABLES: These form the major part of the installation and require attention from the very beginning.

The nameplate particulars of the generators, motors and transformers are noted as soon as possible and the current ratings compared against the cables to be installed.

The types of cables being fitted are also checked particularly to see if any flame extending cables are being used as the Rules have special requirements for these.

The cable runs are examined to make sure that those having different permitted conductor temperatures (e.g., butyl and P.V.C.) are not clipped together or run in the same pipe.

Where P.V.C. cables are fitted, they are checked to see that they do not go through watertight bulkheads or used with equipment where watertight glands are required and these restrictions generally mean that these cables can only be installed in accommodation or similar spaces.

Duplicate feeder cables to the steering gear motors are examined to see that they have not been run together, and it is usual practice on cargo ships to run one set through the shaft tunnel and the other set through the holds or on deck.

CABLE SUPPORTS: All runs are examined to see that the cables are properly clipped and kept as far away as possible from heat sources. They should also be reasonably accessible for examination and it is good practice to limit the number of layers to two.

A sound method of securing cables, particularly those having large cross-sectional areas, is to run them on cross straps welded to an angle iron framework which is supported on hangers. These straps are generally provided with slotted holes as this simplifies clipping.

This method is often employed for the engine room and also for the 'tween deck spaces as shown in Fig. 14, the latter usually being covered in with

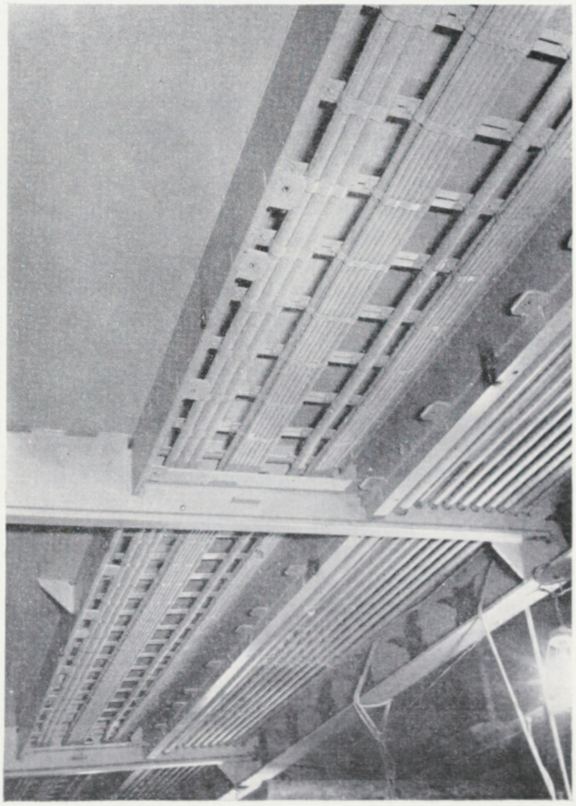


FIG. 14

Cables through 'tween decks

metal covers. With this method a built-up tier construction can be obtained.

Hangers and wooden cleats are also often used for the larger cables in preference to the normal perforated tray plates.

It is the practice in some countries to lay cables in metal troughs without clipping them but this is not considered to be satisfactory by the Author due to the danger of the cables subsequently moving and chafing in service.

Clipping armoured cables direct to metal bulkheads should be avoided, if possible, so that the back of the cables may be preserved through painting.

ACCOMMODATION CABLES: A practice gaining popularity to-day is to install the cables behind the panelling or ceilings in P.V.C. or flexible metal conduits. A complete system is not employed, but rather the short runs to the lighting fittings and switches are so installed. This makes future replacement much easier as the panelling need not be removed. Rubber or P.V.C. insulated and sheathed cables are generally used for this purpose.

Wooden capping and casing are still used but only sheathed cables are now permitted. Care should be taken to see that the cables are securely held in position and the capping should not be used to do this.

Where the cables are clipped to wooden grounds or tray plates, the panelling should be easily detachable, preferably through brass screws.

Aluminium superstructures are often built nowadays, and if any steel cable trays are used, they should be insulated from them with bituminised paper or other insulating materials, or the contact surfaces coated with zinc chromate paint, so as to prevent any possible chemical interaction.

CABLE PIPE SYSTEMS: These are most often installed on the fore and after gangways of tankers or on the open deck of ore carriers, and a typical example of the latter is shown in Fig. 15. Heavy gauge solid drawn pipes, preferably galvanised, should be used as these have to stand up to very heavy weather. Large draw-in boxes are generally necessary and the system properly ventilated to prevent condensation taking place. Draining holes should also be provided in the draw-in boxes, and these holes fitted with a screwed cover.

CABLES ON TANKER GANGWAYS: It is also the practice of some shipbuilders to take the gangway cable pipes through the cargo pump rooms (e.g., T2 Tankers), and where this is done, the pipe wall thicknesses should be at least the same as the pump room wall thickness. Expansion glands are required at strategic points along the cable run (e.g., at draw-in boxes, fore and after bulkheads, pump rooms or at other expansion points) to allow for pipe movement.

Cables without a lead sheath but having a wire braid may now be installed in these pipes. Many owners, however, prefer to install cables on supports so that they may be easily checked and these should, therefore, be protected with cover plates unless lead armoured cables are employed. As the armouring or braiding very easily corrodes, particularly in way of the expansion, a P.V.C. outer sheath is very often fitted and this is far better than the overall cotton braid. This latter tends to harden and split after the cable has been

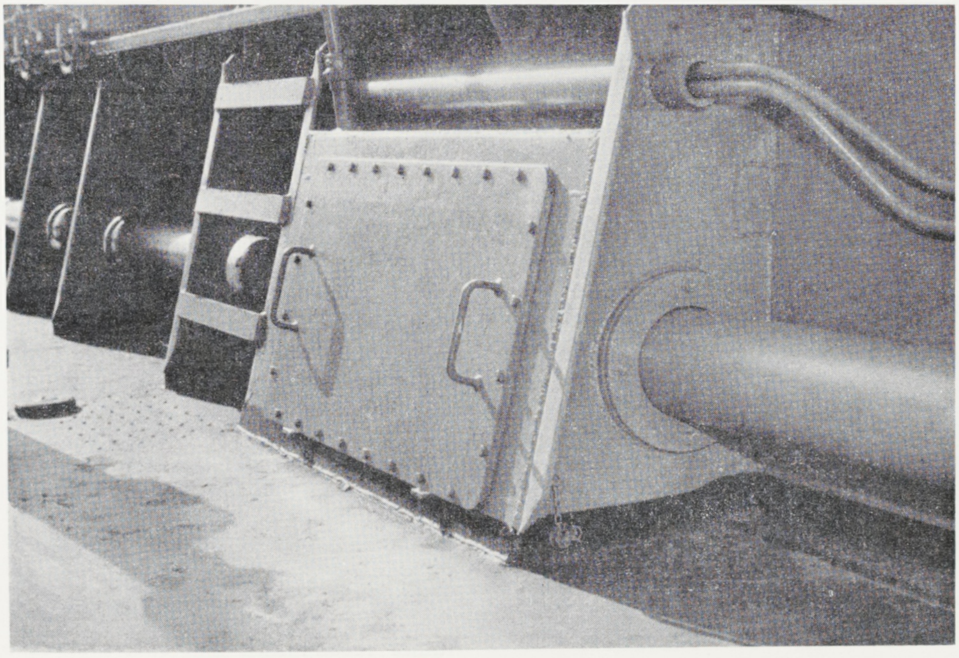


FIG. 15

Cable Pipes along deck (bulk carrier)

exposed to the weather for some time. It thus becomes a water trap and accelerates the corrosion of the armouring underneath.

The cable expansion arrangements are carefully checked to see that the cables or pipes are free to move, as it is not unknown for enthusiastic welders to weld the separate parts together.

CABLES IN REFRIGERATED SPACES: Cables in refrigerated spaces require special attention. When a cable run is to be built into the insulation, the refrigeration Surveyor should be consulted regarding the consequent decrease in wall insulation. Hard asbestos or wood should be fitted between the cable tray supports and the structural steelwork to prevent a possible heat transfer and all metal parts should be galvanised. The sides of the cable run should be closed in, so that it cannot be used for hanging cargo, and where cover plates are required, perforated plates are preferred by the Author as the cables may then at all times be examined.

CABLES THROUGH BULKHEADS: Where cables pass through non-watertight steel bulkheads or structural steel, the holes should be bushed with lead or fitted with a welded metal collar or the cables installed so that they are clear of the hole by at least $\frac{1}{4}$ in. In accommodation spaces wooden or fibre bushes for individual cables are often fitted and care should be taken to see that they fit tightly into the holes.

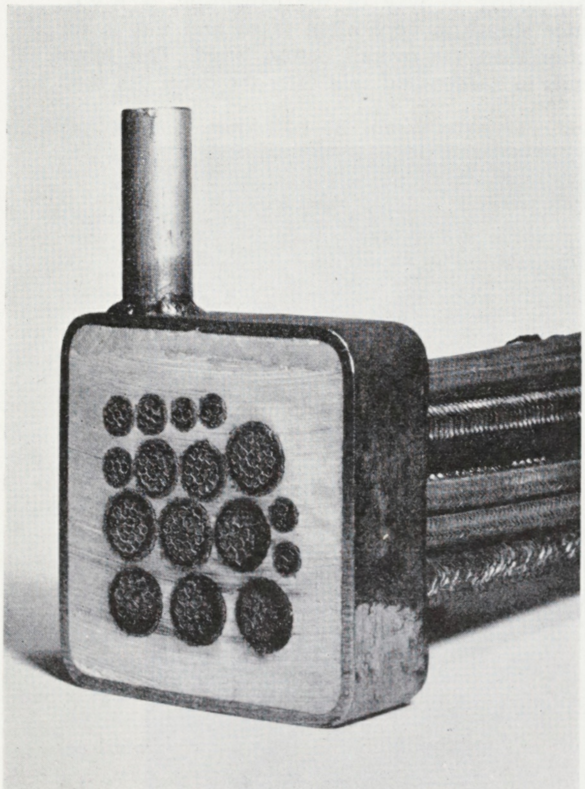


FIG. 16

Bulkhead Cable Box with compound

With watertight bulkheads and deckheads the cables are usually passed through individually packed glands or compounded boxes, and modern practice on the Continent with the latter, is to use an epoxy or polyester based compound. The box and the cables are carefully prepared beforehand, as the compound is generally so thin that it will run out under the cable armouring or braiding, unless they are properly sealed. The box should also be provided with an air pipe to prevent air locks, and both air and filling pipes should finish flush. A typical example is shown in Fig. 16. Once the compound has hardened, it becomes absolutely watertight, and it can also be selected to be oil and fire resisting.

SINGLE POLE INSTALLATIONS: As these are now only permitted with special approval, care should be taken to see that this has been obtained. The connections to the hull should preferably be of brass and be easily accessible. It is, therefore, preferred that in accommodation spaces, where false ceilings are fitted, that the return wire for lighting and small power circuits be brought out to a central point in the alleyways or better still, to the distribution board itself. To prevent any possible magnetic interference, it is considered that the installation should be double pole within a minimum of a 15 ft. radius of the steering compass.

LOCATION AND PROTECTION OF EQUIPMENT: The equipment is regularly checked during the fitting out to make sure that the motors and starters, etc., will be adequately ventilated and suitably located or protected against possible drip or high temperatures on completion of installation. A word at an early date with the Engineer Surveyor about the tank sounding pipes has paid the Author on many occasions as these are very often fitted near motors or generators.

The ventilating trunks, particularly at the switchboards and generators are also examined to ensure that there is no danger of sea water or rain running down them on to the item concerned.

The ventilation of the various small compartments containing electrical equipment are also checked to make sure that an exhaust and supply vent is provided, as, for proper circulation, both are required.

Cable entries are given special attention since where cable glands are not fitted, the cables should enter from below to prevent any possibility of water or oil running down them into the apparatus.

Bulkhead mounted dead front switchboards are checked to see that they can be serviced from the front. The distribution boards enclosures in the machinery and boiler spaces are examined as it is common Continental practice to allow the heads of the cartridge type fuses to project through the front covers. Where such boards are found, the Author recommends that covers or canopies be fitted.

EARTHING: All the cable lead sheaths and armouring or metal braiding require to be

properly earthed. With lead sheaths this is normally done by clips or cable glands, containing a special earthing pressure washer or lead wool or by soldering an earthing wire to the sheath itself. This latter practice should not be encouraged, as damage could result to the cable when too much heat is applied.

The metallic braid or armour is in most cases continuously earthed by being clipped to the structural steel work or cable supports and does not require further earthing. This is shown in Fig. 17. However, some means are generally necessary to keep the spiral armouring tight at the ends and jubilee clips, special armour clamps or glands are usually employed to do this.

Sub-circuit lead sheathed cables need only be earthed at one end and where this is done in the distribution board or some other item of equipment, care must be taken to see that this apparatus itself is earthed as, in accommodation spaces, these are often mounted on wooden or even insulation bulkheads.

Fan motors are frequently fitted to rubber anti-vibratory mountings and these, therefore, must be separately earthed. In cabins, a continuous earth wire looping between the metallic lighting fittings, heaters, and socket outlets is sometimes fitted instead of earthing these items individually.

CABLE TERMINATIONS: The ends of hygroscopic cables such as V.C. are carefully checked to see that they have been properly sealed. The conductor should be soldered or fitted with a cable shoe or similar sealing attachment and then the end taped up. It is also recommended that the taped up end be afterwards painted over with non-hardening varnish.

This latter practice can be applied with profit to rubber insulated cables also as this excludes air from the insulation and prolongs its life. P.V.C. tapes should not be used with rubber insulated cables for the reasons previously mentioned.

SWITCHBOARDS: The circuit breakers and fuses and also the cables are checked against the particulars already noted in the F.E. booklet. There is no easy way of doing this latter, although on the Continent, the cable shoes are often stamped with their sizes. Experience and a good eye also play a part, particularly as one becomes familiar with the standards to which the cables are manufactured. Thus, for example, the largest two-core cable made to the German Marine Cable Standard is 25 sq. mm., and this is helpful when the cable in doubt is supposed to be 35 sq. mm.

It is the Author's practice to carry a small gauge about with him and measure the external diameters of any doubtful cables. These are then checked later against the appropriate standard.

The electrical arrangements are also checked, particularly the preference tripping circuits where group tripping is employed.

Attention is also given to the attachment of the generator and distribution cables to make sure that when these are brought inside the switchboards the clearance and creepage distances between live parts and earth remain adequate.

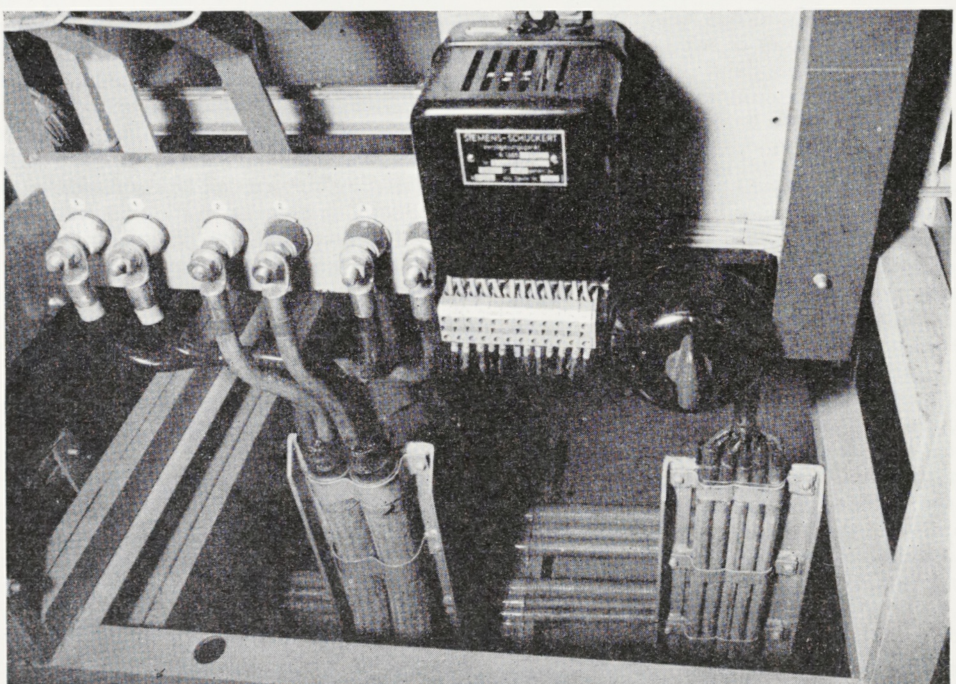


FIG. 17

Cable Connections. Back of Board

LIGHTING TRANSFORMERS: With A.C. installations the lighting transformers are checked to see that with a failure in one unit, there is sufficient reserve capacity to supply the essential lighting. This latter is considered to be the machinery spaces and alleyway lighting, and also the navigation lights. Where single phase units are fitted, it is not necessary in the Author's opinion to connect the spare to the switchboard. It should, however, be located next to the working units and provided with flexible cable connections, so that it may be temporarily connected until it can be moved to replace the damaged unit.

MOTORS: All motor enclosures should be drip-proof as a minimum, and modern practice is to make them of an enclosed, ventilated and drip-proof type. Their location should be checked to make sure that they are protected against drip and splash from pipe flanges and tank sounding pipes.

In particular, refrigerated cargo fan motors should be examined to see that they can, under all circumstances, be easily maintained and removed or repaired in position.

The selection of the motor type is largely dictated by the nature of the drive and the available generating capacity.

(*a*) DIRECT CURRENT: For normal auxiliaries such as pumps and fans, whether either constant speed at varying loads or a range of speeds are required, shunt wound motors are normally used. These are often provided with a light series field to increase the starting torque and to cause the speed to drop off a little under heavy load.

With traction type of drives such as winches or cranes, the series motor is selected because of its high starting torque and quick reversing abilities. The speed is determined by the load and as this can be excessive on light load, a light shunt field is usually provided to prevent this.

(*b*) ALTERNATING CURRENT: The speed of an A.C. induction motor cannot be readily varied as it is dependent on the supply frequency and the number of stator winding poles, both of which are normally constant. Variable speeds can, however, be obtained through special construction (e.g., change-pole motors, wound rotor slip ring motors).

This feature, amongst others, has in the past tended to discourage the adoption of the squirrel cage motor for marine use, but experience now shows that, apart from particular applications, variable speed drives are not necessary.

The squirrel cage motor is used when high starting torques are not required such as with pumps and fans. Where high starting torques are, however, required, wound rotor motors are generally fitted.

STARTERS AND CONTROL GEAR: The enclosures are checked to see that they are drip-proof and that overload and no-voltage protection, where required, has been fitted and properly set for the motor duty. Indicating lights are often specified, and where the main fuses are bigger than 25 amperes, separate fuses should be fitted for these. Sometimes these lights are supplied from an external source and care should be taken to see that this supply is clearly labelled on the starter or connected through the starter isolator.

The starter and control arrangements are selected to meet the motor duty and are generally as follows:—

(*a*) STARTING AND CONTROL

(i) DIRECT CURRENT: All starters are provided with resistances which are switched into the motor armature circuit on starting. The starting steps are designed to give a smooth acceleration and to limit the starting current.

With shunt motors, where speed control is required, this is done by inserting resistances into the shunt field or armature circuits or by a combination of both.

Speed control of series motors is carried out by either switching in series resistances or by providing an armature shunt resistance. For quick reversing, the motor is very often "plugged", i.e., by connecting the armature for reversed connection while the motor is running in the forward direction. In such cases, it is necessary to provide an additional resistance step to limit the inrush current, caused by the increased voltage, to a safe value.

All these resistances may be incorporated in the starters which can be hand or contactor operated and where these latter are employed, the contactor panels are often fitted away from the motors (e.g., cargo winch contactor equipment).

(ii) ALTERNATING CURRENT: The deciding features are the permissible starting current and the required starting torques, and these are considered when the motor is selected.

Squirrel cage motors started "direct-on-line" can take up to eight times full-load current, and if the generators are small, this can be critical. The starting current can be decreased by applying a reduced voltage at starting such as with star/delta or auto-transformer starters, but these have the disadvantage that the starting torque is also reduced. If low starting currents and high starting torques are required, it is normal to provide a wound rotor motor, where resistances are connected into rotor circuit on starting and slowly removed as the motor speeds up.

Although speed regulation of the squirrel cage motor is not practical, it is possible to get two, three or even four different constant speeds by special arrangement of the stator windings. The ends of the stator windings are brought out and connected to a controller so that the number of poles may be changed. Where variable speeds are required, the wound rotor motor with variable external rotor resistances can be used, but this method is wasteful because the resistances are continually in circuit.

As in the case of D.C. machines, the starters may be hand or contactor operated.

(*b*) OVERLOADS: These are either magnetic or thermal devices. Where magnetic overloads are

used, they should either be bridged during the starting period due to the high currents or provided with a time delay device such as a clockwork attachment or an oil dashpot.

Thermal overloads are preferred by many engineers because of their inverse time characteristics, but they should also be out of service during starting periods where these are relatively long, as for example, with separator motors.

If proper protection is to be afforded to the motor, the overload setting has to be carefully selected, particularly to-day where the motors are generally rated with a limited overload capacity (e.g., 50 per cent for 15 secs.). It is normal practice in Germany to set the thermal overload at full load, as it will allow the motor to carry an overload of approximately 5 per cent for over two hours. With a heavier overload, the corresponding operating time is decreased.

This selective form of protection cannot be provided by a magnetic overload which has no damping device and in such cases it is normally set at full load or about 5 per cent overload.

Protecting intermittently rated, change-pole A.C. motors such as windlass motors is a different and far more difficult problem, due to their duty and to the thermal effects of the starting currents. Thermocouples attached to the windings and connected in the tripping or under-voltage release circuit of the motor contactor are, however, an effective way of doing this.

Steering gear motors should not be fitted with overload protection, short circuit protection only being required.

(*c*) No-Voltage Releases: Motor starters should also be provided with a no-voltage release to prevent the motors restarting after a voltage supply failure as this could be dangerous to plant and personnel. Automatic restart, however, is sometimes required as for instance with steering gear motors, domestic sanitary and fresh water pumps or fan motors and the no-volt release need not, therefore, be fitted for these drives.

The no-voltage release feature is normally an inherent part of the starter as a voltage coil is generally fitted to hold the starter arm in position or keep the contactor closed.

Transistor Control: The advances made in transistor techniques have resulted in their being introduced on board for steering gear control circuits. With such an essential auxiliary, however, an alternative control circuit should, in the Author's opinion, be provided.

Remote Stops: The connections adopted for the remote stops of fans and oil pumps should be checked to see that the principle of "failure to safety" has been adopted, and by this is meant that the tripping circuit does not depend on a coil being energized. If this were so, the tripping circuit could be inoperative through the coil being burnt out without the engine room staff being aware of the fault.

The normal practice is to connect the switch or push button in the under-voltage or holding coil circuit of the starter or contactor.

As the stops for the engine and boiler room fans, O.F. and cargo pumps must be fitted outside the space in which they are located they are generally positioned in the engineers' alleyway.

Lighting Fittings: Due to their higher efficiency, fluorescent lighting fittings in machinery spaces are frequently seen to-day even with D.C. installations. Polyester and glass fibre materials are often used for the enclosures, and these are also acceptable for tankers, being approved as required in M 1612.

All lighting fittings, however, require watching with regard to the internal temperatures, as these can be very high indeed. Fluorescent fittings are generally in order, although the starters and chokes can also be very warm. The permissible temperature inside the fitting is limited to the temperature that the connecting cable can withstand.

For rubber and P.V.C. cables this is 60° C. and based on an ambient temperature of 45° C. in all spaces, except accommodation, where 25° C. is reasonable, this allows a temperature rise in the fitting of 15° C. and 35° C. respectively.

However, with the weatherproof tungsten lighting fittings normally used, the temperature rises are often much more than this. Some manufacturers produce fittings with attached cable connecting chambers, insulated from the fitting, but in Germany for instance this is not so and the practice there is to fit separate metal clad junction boxes adjacent to the fitting and connect them with silicone cables.

The same basic arrangement is also adopted in the accommodation spaces, where decorative fittings, selected either by the architect or the owners, are used, the junction boxes however being made from porcelain or plastic materials.

Dangerous Spaces: The electrical equipment on ships carrying dangerous liquid cargoes in bulk deserve special mention. These cargoes can be liquefied gas under pressure, low flash fuel oil or even wine.

The ship which requires particular attention is the combined ore carrier/tanker, as these present many problems. They are checked to see that normal working spaces such as the alleyways between the ore holds and side oil tanks are safe spaces, so obtained through cofferdams. If these precautions are not taken, the alleyways themselves become cofferdams in which no electrical equipment, not even lighting fittings, are allowed. The same applies to the midships accommodation space, which often sits on top of a joining alleyway.

Similarly, the ventilating fan motors for the ore holds, even if flameproof, should not be fitted in the ventilation trunks and the same applies to the tanker pump room fans.

If the E.S.D. oscillators are fitted in the forward pump room or cofferdam, the enclosure is examined to see that it has not a common bulkhead with a cargo tank.

It should also be noted that a 'tween deck space can be the forecastle, as sometimes the cargo

tanks extend under this space and this is always checked at an early date.

The selection of the correct enclosure to be used in the dangerous spaces is a fairly simple matter in the United Kingdom as, apart from intrinsically safe apparatus, there is only the flameproof enclosure, grouped in accordance with the various gases.

On the Continent, however, there are very many different enclosures, all of which are called explosion proof, ranging from the Ex(s) "special arrangements" to the Ex(d) "pressure tight". Only the latter is permitted and this must also be selected for the correct gas group.

The designation Ex is generally cast into the enclosure in much the same way as the FLP mark but the type can only be checked from the name-plate which is always fitted.

A typical Ex(d) lighting fitting is shown in Fig. 18. In Germany these fittings are always provided with a double-pole switch, built into the main housing and interlocked so that the fitting is made dead before the glass can be removed for lamp renewal. The cable terminal box is also Ex(d). The Ex(e) fitting, which was the subject of Circular 2171, is shown in Fig. 19.

Cargo oil tanks are very often gas freed to-day by using portable motor driven fans. The motors are flameproof or Ex(d) and are supplied from contactors and plug sockets fitted in watertight boxes built on to the gangways or superstructure. A typical arrangement is shown in Fig. 20. Unless the control gear is also flameproof, it must be located at least 10 ft. away from a tank opening or pump room, and the socket should preferably have a built-in switch, so arranged that the plug cannot be withdrawn whilst alive.

Finally, when the installation is practically completed a general examination is made to see if any cables or equipment require protection or if additional supports are necessary to prevent vibration.

(b) Testing

The tests to be carried out are described in the Instructions to Surveyors and need not be repeated here.

It might, however, be of interest to other Surveyors to describe some methods adopted by the Author in carrying these out on A.C. installations.

(i) Overloads and Preference Trips

These are tested by either (*a*) running the alternator in short circuit at reduced excitation, or (*b*) reactive current.

With method (*a*) it is necessary to disconnect the alternator panels from the rest of the main

Fig. 18

Schanzenbach Ex(d) Lighting Fitting

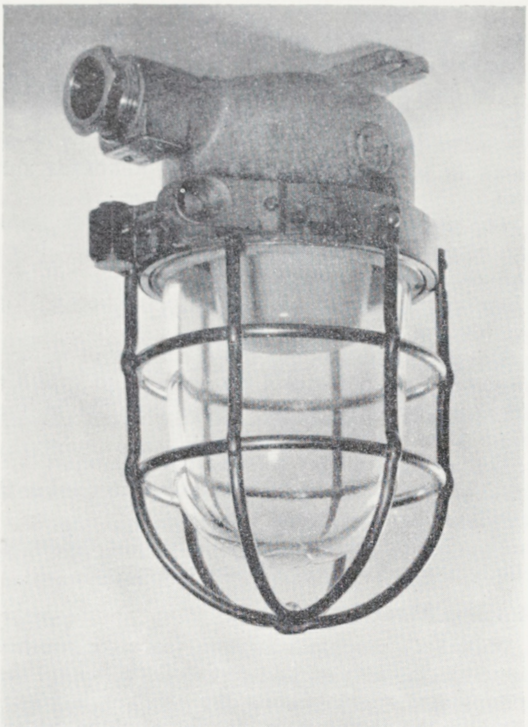


Fig. 19

Ex(e) Lighting Fitting

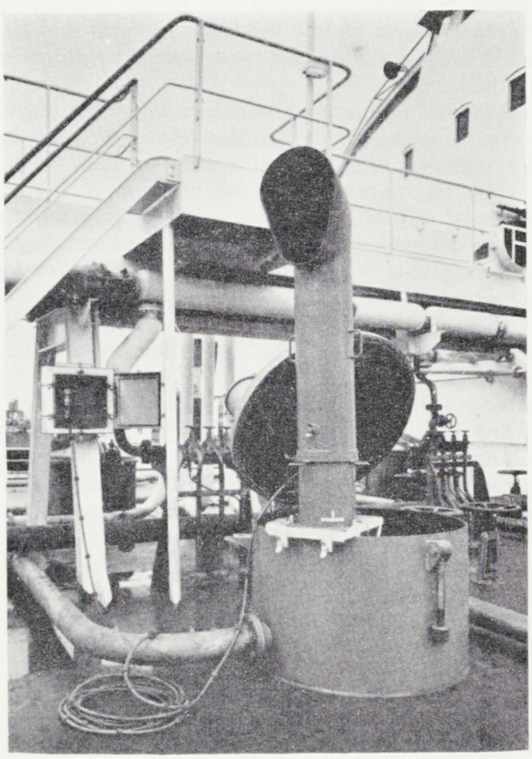


Fig. 20

Siemens Cargo Tank Gas Freeing Portable Fans

switchboard and to separately supply the auxiliary drives essential for the prime movers (e.g., cooling water pumps, condensate pumps, lubricating oil pumps, etc.). It is also better to do this with method (*b*).

With (*a*) the voltage range of the hand regulator is first checked to make sure that the alternator voltage can be reduced to zero. When this has been established, the busbars are short circuited, one alternator connected to them with the voltage reduced to zero and then the excitation slowly increased until the desired current is obtained. With self-regulating alternators without hand regulators, it is necessary to insert a variable resistance into the excitation circuit to do the same work and a suitably selected welding transformer is excellent for this purpose and has been used by the Author on many occasions. This method is particularly advantageous when thermal overloads are fitted, since overcurrent as required can be applied without overloading the prime mover.

This method (*a*) can also be used for testing the feeder circuit breaker overloads by placing the short circuit at the circuit breaker outgoing terminals.

With (*b*) two alternators are operated in parallel and the excitation of one either reduced or removed, depending on the circuit arrangements, causing it to draw wattless current, which can be used to test the overloads.

(ii) Voltage Regulation

Difficulty is often experienced in obtaining the required lagging power factor load. However, an ohmic load, such as a water tank or ballast resistances combined with that obtained from running the ship's auxiliary motors at no-load, generally gives the load required, as these motors, particularly deck A.C. auxiliaries, run at a very low power factor on no-load.

(iii) Reverse Power Relays of Turbo-Alternator Sets

As stated in the Instructions to Surveyors, these should be set to operate between 2–4 per cent. This is because the amount of power required to drive a turbine operating in a vacuum is very small. These are, therefore, best tested under working conditions by shutting off the steam from the turbine so as to make sure that the setting is not too high. They should also not be set too low, otherwise difficulty will be experienced with synchronising. A time delay of approximately two seconds for these relays is therefore recommended by the Author for this reason.

(iv) Short-Circuit Trips

These are tested by hand to make sure that they operate instantaneously.

(v) Discrimination

It is sometimes necessary to make sure that the protective devices have been correctly selected and set and the Author has from time to time tested that the alternator circuit breakers have

stayed in, when a short circuit has taken place on an outgoing feeder. A main or auxiliary switchboard switch and fuse (and not circuit breaker) have been short circuited to prove this.

(vi) Synchronising Arrangements

Where semi-automatic devices using reactors are fitted it is necessary to ensure that the alternators can also be synchronised when the chokes are, for some reason, out of service. Thus with the circuit shown in Fig. 12 the synchronising is carried out using the single pilot lamp.

The following tests, not mentioned in the Instructions to Surveyors, are also always witnessed by the Author.

(vii) Remote stops for the fans, oil fuel and cargo pump motors.

(viii) Navigation Indicator Board.

(III) PERIODIC SURVEYS

The Instructions to Surveyors detail the faults to look for but there are several additional points which the Author would like to mention.

It is his experience that a routine approach to every Survey simplifies the work for the Surveyor and the practice adopted by him is to start forward, work aft and then finish off in the engine room.

The windlass motor and control gear are first examined and then the lighting fittings and cables in the paint and lamp rooms are inspected. The forecastle space and small stores normally found forward are also examined, as the electrical equipment here is sometimes subjected to abuse.

The fittings, cables and equipment in the deck houses are then checked and the next move is into the 'tween deck spaces for the main cable runs and lighting fittings, as these are very prone to mechanical damage.

If the ship is a tanker, the gangway cables are inspected, particularly in way of the expansions. If armoured and braided cables have been fitted and the braiding is defective, it is well worth suggesting to the owners' representative that this be removed as it very often is a water trap and consequently accelerates the deterioration of the armouring. The expansion arrangements are also checked to see that they are still operative. The pump room lighting fittings are then examined and if the E.S.D. oscillators are located there, the enclosure is always checked to see that it is still tight.

The 'tween deck spaces on tankers are given special attention as switches and sockets are often found in the workshops and offices sometimes situated in these places.

Having completed the inspection of the deck equipment and cables, the steering gear is examined and this is followed by a walk through the accommodation spaces ending up in the wheelhouse. During this examination any temporary lighting fittings, portable heaters, hot plates, radios, fans, etc., and associated cables are noted and the Superintendent advised that they should either be removed or permanently installed.

The navigation indicator board in the wheelhouse is examined particularly to see that the visual or audible alarms are in order.

The battery compartment is also normally situated on this deck and this is checked to see if the batteries are in order as if damaged this could result in acid spilling against the bulkheads and deck.

The accommodation distribution boards fuses are often found to be faulty, i.e., broken fuses and bad contact between fuse and fuse holders, These circuits should be particularly checked for overloading.

The engine and boiler room equipment is then inspected and the cables for gauge and cylinder lighting circuits and also those installed in the bilges given special attention.

The cables in the refrigerated chambers are checked to see that they have not been used as a casual means of hanging cargo.

The Author has also found that the following points are well worth particular attention.

Rotating Machinery

With D.C. machines, carbon from the brushes compounded with oil and dirt, collects behind the commutator risers because the air flow is generally over the commutator into the machine. Sometimes the carbon also lodges between the insulation under the commutator and the "V" ring and in such cases the insulation resistance is generally zero. To repair this, the commutator has to be stripped and as it cannot be done on board, it should be removed ashore to a proper workshop. Another source of low insulation resistance is the brush arm insulation and these should be properly cleaned.

With self regulating alternators where the transformers, chokes and rectifiers, etc., are fitted to the machine, the internal wiring of this device should be examined for overheating. If P.V.C. insulated cables have been used and are tightly strapped together, the cables should be examined at these points of pressure to see if any conductors are exposed.

The slip rings and connections should also be examined especially the insulation where the cables pass through a slip ring of opposite polarity or where they emerge from a hollow shaft.

The D.C. field coil insulation of synchronous motors, which are also provided with short circuited rotor bars for starting purposes (e.g., propulsion motors), should be examined to see if it is damaged as these bars generate high temperatures on starting. The stator slot wedges should also be checked for slackness and any tell-tale powder at the slot sides generally indicates that they move in service.

Cable Ends

Where these are found to be in a poor condition, they can often be repaired by sleeving them with insulation tubes. Silicone cable sheaths are quite good for this purpose but they should not be used where likely to come in contact with sharp edges or corners.

It is good practice even if the cables are in good condition to paint them with a non-hardening varnish or resin. Shellac in the Author's opinion should not be used as this dries hard and the next time the cable tail is moved for a megger test, the insulation might crack.

Electro-Magnetic Slip Coupling

The air gaps should be carefully checked as the out-of-balance forces involved when these are unequal can be quite considerable. They should however be checked after all the machinery involved has reached operating temperature. The results obtained should then be compared with readings previously taken and the manufacturers' manual examined for information regarding the maximum tolerances permitted.

Repairs to Machines

Where windings are part renewed, the new windings should be given a full high voltage test before connection to the existing windings.

Testing

The generators should be tested with ship's load for governing and compounding/regulation and also for parallel operation and synchronising. The reverse power/current relay can be checked in the usual way, while the overcurrent, preference and short circuit releases can be tripped by hand to show that the tripping mechanisms are in order. With A.C. generators, it is often possible to test the overcurrent trips with reactive current as previously explained.

The Author also tests the remote stops for the pumps and fans and the operation of the navigation indicator board.

Finally, the steering gear and windlass equipment is generally inspected to see that they operate satisfactorily.

CONCLUSION

The foregoing remarks express the Author's personal opinion and if it appears that emphasis has been placed on some aspects of the installation while others have been omitted or only received casual mention, this is because these have generally presented him with the most problems. It is hoped, however, that this will be rectified in the forthcoming discussion.

The Author wishes to express his grateful thanks to the following firms for permission to reproduce their diagrams, graphs and illustrations:

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Discussion

on

Mr. W. Morris's Paper

THE SURVEY AND TESTING OF MARINE ELECTRICAL EQUIPMENT

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

Discussion on

Mr. W. Morris's Paper

The Survey and Testing of Marine Electrical Equipment

Mr. C. HAFFNER

Perhaps one of the most difficult things to do when writing a paper of this kind is to know where to start and where to finish. The survey of electrical equipment has so many facets that it is almost impossible to cover all of them adequately. As the Author has pointed out at the end of his paper, he has had to select those which might present special problems, and I think he is to be congratulated on his choice of subjects, which has produced some observations which should be very useful to other Surveyors who are confronted by similar tasks.

The Rules for Electrical Equipment are intended to provide guidance for Shipbuilders, Manufacturers and Surveyors in producing a reliable and easily handled source of electrical supply for all kinds of services on all manner of ships, and do not pretend to replace the itemised builder's specification which should be the basis of manufacture and installation. Inevitably, the Surveyor is sure to be faced from time to time with decisions to make concerning interpretation of the Rules and, although the Instructions for Surveyors are intended to assist him in this, this paper should also help by describing the methods used by an experienced colleague.

In connection with the issuing of certificates in respect of survey of equipment which is to be examined and tested at the makers' works, an official notice is now being prepared, dealing with the subject very much along the lines suggested by the Author under the heading, "Factory Work" on page 1.

Commutators

It would be appreciated if the Author would expand his remarks concerning commutators with pressure moulded insulation, and if he would say whether the movement of parts occurred in service or if this fault became obvious during assembly or test. This disability was foreseen and commented upon when this form of construction was investigated some years ago at Headquarters.

In the section dealing with the provision of mica or equally effective material between windings and core or frame of machines otherwise employing Class A or E insulation, polyester film could be added to the list of materials acceptable for this purpose.

Temperature Tests

Perhaps it should be made clear that the pad of thermal insulation to be placed over the bulb of a thermometer used for measuring the temperature rise of machines under test should be a small one. Its function is to prevent loss of heat from the bulb, not from the object under test, and a piece which covers a large area could cause false readings to be recorded.

Voltage Regulation Tests

When carrying out voltage regulation tests on a generator, it might be advisable first to attempt to obtain particulars regarding the governor characteristics of the actual engine to be coupled to it, and to match the speed of the driving motor to this, where the generator is being tested apart from its prime mover. In the absence of precise details, the figure of 4 per cent is perhaps a reasonably average one to adopt.

Switchgear

In discussing switchboards, the Author states that although the survey of equipment, other than that connected with electric propulsion, is not a Rule requirement, this is often done by request. It could be added that whether a request is made or not, inspection during manufacture and assembly is usually rewarding, as many errors can be corrected at that stage with the least trouble. If it is reasonably convenient to do so, it is suggested that a visit or two should be arranged.

It should be noted in passing that, although a list of approved circuit breakers is in prospect, the preparatory work is not yet complete. The current Rules do not require the formal approval of circuit breakers, which at present may be accepted by the Surveyors on satisfactory evidence of their suitability. When the list is published it should furnish all necessary particulars, so that circuit breakers as well as fuses can readily be selected for the appropriate duty.

Insulating materials for switchboards should have one property in addition to those mentioned by the Author. Considerable force is involved in the operation of some switchgear, and the fractures which sometimes occur would be avoided if attention were paid to the mechanical strength of these materials.

Circuit Breakers

Will the Author explain his statement that all feeder interrupting devices at the main switchboard should have equal interrupting capacities?

It has occurred that preference tripping circuits have been fitted with switches in order to cut out the tripping circuit of individual units or sections. This is of course undesirable, as this form of protection is intended to be entirely automatic. It was thought well to mention this because, although seldom encountered nowadays, such switches were discovered by a Surveyor recently, fitted to a new installation under construction.

The provision of adjustable current settings for preference tripping devices, mentioned in the revised Rules, was not intended to bar time-based settings, as implied by the Author. It would be interesting to hear if it is the practice in Germany to use preference tripping based on progressive current settings, and, if so, how it is employed, and why it is preferred to a time-based system.

Cable Supports

While indiscriminate laying of cables in troughs without further means of fixing in position is in general deprecated, some shipbuilders, by paying particular attention to dimensions and ensuring that the cables fit snugly in the troughs, have dispensed with clipping quite successfully. In general, however, the Author's remarks are concurred with.

It would be interesting to hear some further particulars as to how the builders deal with the problem of fixing steel tray to aluminium superstructures. In particular, what material is used for the bolts?

Cables in Refrigerated Spaces

In connection with cables in refrigerated spaces it should be noted that in general cables should not be buried directly in thermal insulation, but if it is necessary to have the cables below the surface level they should be contained in a trough of some description, such as that described in this paper.

When cables are passed through a fireproof bulkhead, additional fireproof lagging may be necessary, over and above the resin compound mentioned in connection with Fig. 16.

Switchboards

The Author might be invited to comment on his experience with the shipbuilders' choice of location for the main switchboard. This long-suffering, but essential, part of the installation seems to have a fatal attraction for pipe flanges, oil separators and the like, so that a sharp eye has to be kept on the possible encroachment of such enemies, which are capable of directing sprays of oil, water or oil vapour on to the switchboard equipment. It might be added here that it is not only the oil itself which constitutes a danger; an oil film collects dirt which can cause tracking and ultimate breakdown of the insulation. A favourable position, even for deadfront switchboards, is most desirable, and early intervention can sometimes prevent mistakes in this direction.

Dangerous Spaces

At the foot of page 17, the Author mentions a forecastle construction in which the enclosed space extends over the cargo tanks. As he points out, this space is subject to the same restrictions on electrical equipment as that in the centrecastle. Too much attention cannot be given to early confirmation that the builders who employ such a construction appreciate the position, as great difficulty attends any rectification when construction of the ship is well advanced. It is seldom obvious from the plans alone, the approval of the several sections of which are dealt with in different departments, that it is proposed to install unsuitable electrical equipment in such spaces.

Discrimination

The Author might be requested to explain why he excludes circuit breakers from his test for discrimination.

Mr. F. B. MORT

Mr. Morris must be congratulated on his paper which I am sure will be used for reference by many of our colleagues who are not specialists in the electrical field. To be able to cover the wide range of equipment and problems involved during the survey of electrical installations of modern ships in one paper is extremely difficult and I think you will agree this has been done admirably by the Author in a very concise manner.

The Author has indicated his ideas on the protection of motors and I think this feature should be emphasised in view of the relaxation by the Rules which now permit the use of continuously maximum rated machines, that is, motors or generators which have no sustained overload capacity at all, whereas in the previous Rules rotating machinery was required to have a sustained overload capacity of 25 per cent.

It is especially important with A.C. induction motors in order to avoid damage to these machines caused by relatively marginal overloads that the protective devices be carefully set so that they will operate when a sustained overload of 5 to 10 per cent occurs.

As mentioned by the Author, however, there are certain essential motor drives such as steering gear, where it is considered prudent to risk damage to the motor in order to maintain the service. Control gear for these motors should have an overload relay set to operate as indicated previously but arranged to give us alarm instead of tripping off the motor.

The Author states that the type of A.C. switchboard favoured on the Continent has fixed type circuit breakers with, if specified, isolating links or switches between the breaker and the bus bars, rather than withdrawable type breakers as used in normal British practice. I have recently had the opportunity of examining some A.C. switchboards built on the Continent and it appeared to me that to carry out maintenance or adjustment, on the fixed type breakers installed on these boards, even though isolators were fitted, could be a hazardous job and I should be glad if the Author could state the reason why this practice is favoured on the Continent. These remarks do not apply, of course, to the moulded case type breakers which are normally grouped in separate sections of the switchboards and backed up by a large breaker or fuses although some manufacturers are now producing moulded case circuit breakers which can be plugged in rather than permanently fixed.

Mr. Fennell

I would like to congratulate Mr. Morris on his presentation of such an interesting paper. The subject covered is so wide that a volume could be written on it, but he has been able "to get at the meat of the subject" in a very concise paper.

I would like to comment on the following points: —

1. The fitting of "shrink rings" on long commutators has caused quite a deal of trouble, and I consider that mica should be used as insulation under the ring. Some manufacturers use micanite and difficulty then arises due to the spongy nature of this material. The ring may appear tight after fitting, but under a normal operating, heating and cooling cycle it often "slacks off".

A reasonable "rule of thumb" method for the shrink fit is about ·001 in. per inch diameter of commutator.

2. Care should be taken when core bands are fitted on armatures after a repair to make sure that the correct type of wire and securing clips have been used. In many D.C. machines it is necessary to fit non-magnetic banding wire and high resistance metal clips (e.g. Ferrystrip) which has a resistance value of about 12 times that of tinned copper and a tensile strength in the same region.

I know of many cases where this has been overlooked and the bands and clips have run red-hot and have "thrown", resulting in serious damage to the armature and fields.

3. Whilst manufacturers' investigation tests are outside the scope of the paper I think that Surveyors, with the agreement of the manufacturers, should endeavour to see as many of the investigation tests as possible. One can learn a great deal about the operating characteristics of the equipment manufactured in one's own district and this experience and advice can be passed on to colleagues in other ports.

4. I agree wholeheartedly with Mr. Morris that in the case of alternators the voltage regulation tests should be carried out with their associated A.V.R. or static excitation equipment. A great deal of time and inconvenience can be saved aboard the ship by the Surveyors and builders if this equipment is proved before dispatch from the works.

5. The fitting of the fuses after a rotary switch on a main switchboard is a practice to be deprecated. As Mr. Morris says, the condition of this type of switch cannot be checked and a fault on this switch (which is not uncommon) is a bus-bar fault.

This practice is probably a consequence of a previous Rule M 313. Where the switches referred to were open-type knife switches and even in their case it is debatable whether the fuse should be before or after the switch. I think safety of the personnel was the primary consideration when this rule was formulated.

6. An easy method of checking the polarity of the reverse current device is to connect a test lamp across the equaliser and the pole in which the reverse current device is fitted. The test lamp should light, if it doesn't, then the device is in the wrong side.

New Construction

7. One of the most important compartments to examine for adequate ventilation is the battery compartment, and this should not contain any other electrical equipment (e.g. fuseboards, switches, etc.). It should be treated as a dangerous space.

8. I would rather see each cabin, or two cabins, with a separate earth connection to the metal. One earth terminal for a large number of cabins is not, in my opinion, good practice.

9. I agree with Mr. Morris that the fitting of transformer link boxes and a spare single-phase unit permanently connected to it is not necessary. It must be a very costly business to the owner and builder.

10. I don't think it wise to generalise on the fusing of pilot lights, and a 25 amp. fuse, in my opinion, is far too large for pilot light protection. Is this Continental practice?

11. One of the main troubles with the selection of electric motors is that there doesn't seem to be the right liaison between pump, motor and starter designers and manufacturers.

In my opinion, a correct assessment of the mechanical requirements should first be made, for no matter what the motor characteristics are, the driven part will only utilise that part of the motor characteristics it requires, and the starting, accelerating and running torque requirements should be considered before a suitable motor can be chosen to match them.

When this information has been acquired the control gear designers will then have some facts to work on.

We all too often see the results of the mechanical and electrical engineers not co-operating in this field.

Mr. F. H. Tickell

I should like to take this opportunity to congratulate Mr. Morris on his very informative paper upon which I am sure a lot of midnight oil must has been spent.

In the introduction to the paper the Author states that British Standards and Practices have been dropped in favour of I.E.C. Standards. I feel, however, that this statement is rather too sweeping because there are still many British Standard clauses embodied in the New Rules and some of the tables bear a close relationship to British Standards, i.e. B.S.2949, Rotating Electrical Machines for use in Ships.

Core Assemblies and Frames

Mr. Morris has given us a clear picture of the points which the Surveyors should note with care when surveying machines during construction. A further hint which may be of interest concerns the building of armature or rotor cores. I have found that if the end plates carrying the stiffener

risers of the core assembly are given a slight inward set in way of the core teeth, this results in an additional pressure being exerted on the core teeth when the final pressure is applied prior to the locking of the core. An easy way of testing laminations of a core for tightness is by means of a penknife blade, insertion of the blade at various points along the length of the core should not penetrate the laminations by more than one-quarter of an inch when ordinary pressure is applied.

With regard to the welding of arms to armature shafts, the Author might have stressed that no circumferential welding on the armature shaft is permitted. I know that my engineer colleagues are aware of this fact, but perhaps there may be some of my electrical engineer colleagues who have not appreciated this point.

Windings and Insulation

At the present time the Society's rules do not permit the use of polyester resin insulated conductors, i.e. such as "Lewkanex" for use in the windings of rotating parts of machines and I should be glad to hear of any information which the Author may have on this type of insulation which may be in general use on the Continent.

Self-regulating Alternators

A number of firms in the United Kingdom are fitting silicon rectifiers in this type of machine because of the reduction in size for a given output as compared with the selenium type of rectifier, also a higher temperature rise is permitted for the silicon rectifier.

The silicon rectifier is comparatively new for this duty and it would be interesting to hear the Author's remarks on the performance in service of this type of rectifier.

It may be of general interest to Surveyors to note that machines fitted with silicon rectifiers should on no account be subjected to a high voltage test at the makers' works, unless the rectifiers and any feed-back circits have been shorted out, otherwise failure of the rectifier will result. Similarly, during special surveys, the same precautions should be observed when insulation tests by means of a megger tester are carried out.

Compression or Crimped Cable Sockets

I am in full agreement with the Author in his preference for the compression or crimped type of socket as opposed to the soldered type.

The crimped type of cable socket with which I am most conversant leaves little chance for error.

Each size of socket has a number embossed upon it and reference to a list of sizes indicates the correct socket size corresponding to the conductor size to be used. The bench and hand hydraulic tools used for crimping are fitted with interchangeable dies stamped with the socket numbers and once the operation of crimping is commenced the socket cannot be removed from the tool until the correct pressure has been applied. Furthermore, the die embosses the body of the socket with its appropriate number which matches the number already stamped on the socket. This is a check that the correct die has been used. If an incorrect size of cable had been used then the die number will not be reproduced on the socket.

I recall a recent case of alternators fitted with flexible leads between the windings and the terminal boxes and fitted with the soldered type of socket.

Failure of an alternator resulted from inefficient soldering of the cable sockets. Examination showed that the sockets used were too small for the cable conductor size as it was noted that the outer strands of the cable had been cut away. Due to the tight fit of the cable conductor in the socket, the solder had not penetrated the socket and had only flowed around the top of the socket and conductor.

Local heating at the socket occurred with the alternator on load and the cables parted from the sockets thus causing a supply failure.

These alternator cables were subsequently renewed and a compression type of socket fitted.

Transformers

The new Rules require proposals for the use of liquid-cooled transformers to be submitted for consideration.

In view of the use of high voltages for large installations it may be necessary to consider the use of liquid-cooled transformers and it would be interesting if the Author could indicate what he considers would be a suitable type to submit for approval, particularly one fitted with Buckholz protection.

Transistor Control

The Author states that where this type of control is fitted for steering gear controls, an alternative control circuit should be installed. Does this statement imply a duplicate control or another system altogether? What are the objections to transistor controls, if any?

Dangerous Spaces

Reference to explosion-proof enclosures in the paper includes the term "pressure tight" in connection with lighting fittings. I should be glad if the Author would make a comparison between this type of fitting and British Standard 229 flameproof fittings, with reference to service conditions.

In conclusion may I once again thank Mr. Morris for a most interesting paper which will prove of value to all Surveyors who have to carry out Electrical Surveys, both in the factory and on the ship.

Mr. F. P. Crum

I should like to compliment Mr. Morris on his well compiled and very comprehensive paper. The paper will undoubtedly be of great assistance to Engineer Surveyors who have to supervise electrical installations.

The section on testing at manufacturers' works is very interesting to a Surveyor in a district

limited in the number of large electrical manufacturers of rotating machinery. The point to be raised here is the acceptance of works tests for the transient voltage response of self-regulating or A.V.R. controlled alternators. During final tests on board ship, experience points to the voltage control apparatus as being a source of trouble and it is felt that the testing at works may eliminate some of this trouble. While on the subject of self-regulating voltage control, the Author's comment would be appreciated on the procedure of fitting this apparatus with no alternative means of regulation by hand. A case has been experienced where, due to the failure of small rectifiers in the regulator circuit, two out of three alternators installed became unstable and inoperative. The fault was due to the machines being shut down and the regulating circuit not being opened. It appears therefore, that where the effectiveness of an essential control system relies on light pieces of equipment, a more robust, but not necessary more effective, alternative means of control should be provided.

With reference to "Klippon" terminals used with V.C. cables, it has been found that to solder the ends, or fit a crimped sleeve on the cable, does not allow the strands to be compressed sufficiently to give adequate contact area, or a mechanically sound connection and where such cables are used a different form of the same make of terminal is adopted, which allows for crimped or soldered sockets to be used and bolted.

Under the section on A.C. motors there is no mention of the "high torque" type squirrel cage motor, which gives a reasonably low starting current, is more robust, simpler and cheaper than the wound rotor type, and in most instances can be used where starting current must be limited. It has been found that a discussion with the electrical drawing office on starting currents, torques and types of starting is often advantageous at an early stage in the construction.

Mr. J. C. WRIGHT

In view of the recent introduction of new Rules relating to Electrical Installation, Mr. Morris has, I consider, chosen a most opportune time in preparing this paper, which gives an excellent description of the electrical equipment on modern ships and in factories. The items of major importance to check when inspecting machines and equipment in factories has been well taken care of, although I feel that with the normal time that a Surveyor has for inspection work, it would be rather a problem to investigate all the points mentioned.

The reputation of a manufacturer, I feel, should be taken into account. A firm who has been producing machines or equipment for a considerable number of years, which have proved satisfactory, do not, I feel, need to be checked so thoroughly for methods of manufacture. However, the special requirements for marine work which differ from the standard industrial requirements are points I consider require careful attention—insulants and impregnation, etc. The insulation on leads from stator windings to terminal boxes has given trouble in a number of instances. Also on D.C. machines, bonding wires on armatures have burst owing to probably having been soldered with a spirit flux. On D.C. machines it is also important to check that the series coil is connected to the negative pole.

With regard to synchronising arrangements for A.C. machines, I note the Author's remarks regarding the advisability of having an alternative to the synchroscope, e.g. lamps, but I also consider that it is advisable to have a spare voltmeter, as if either the bus bar voltmeter or the incoming machine voltmeter breaks down synchronising would be difficult.

Under the heading "Testing" the methods referred to for overload trips, namely, the shorting of bus bars and reducing the alternator hand regulator to zero—as a matter of interest my records of the ships that I have tested show that the lowest voltages obtainable on 440 volt installations have been from 90 to 350. However, this could be overcome by inserting additional regulator resistance, as mentioned by the Author, for self-regulating alternators.

With regard to "Periodic Surveys", I find that the first and perhaps the most essential item is to discuss with the Chief Engineer or the ship's electrician how the installation has been operating. One can obtain information in this way that is quite impossible to obtain by visual examination. When attending for a survey the ship is normally without its own power, and it is not possible to check how the A.V.R. or the engine governors, etc., are operating, until the ship is preparing for leaving. I have mentioned this point as Surveyors without much experience of survey work may proceed with an examination together with the electrical contractor who is carrying out the repairs. It is also of great importance to leave a list of recommendations with the Owners' Superintendent, or Chief Engineer, and not with an electrical contractor as some recommendations may require explaining to the owners who are responsible for paying the cost of the repairs.

Mr. E. D. COOK

I have enjoyed the paper by Mr. Morris and look forward to having it in more robust covers for future reference. He is to be congratulated on the volume of practical experience which he has introduced.

May I contribute and seek enlightenment in connection with his closing pages Nos. 20 and 21.

Periodic Surveys

Mr. Morris has indeed given us a route to follow when examining electrical equipment, much will be gained by a close study of his procedure. May I add to his comprehensive list.

A close examination of electrical equipment in the tunnel escape, where it is often in the very bad company of steam and water, can be most rewarding.

When testing the steering gear the alternative supply should not be overlooked as it is sometimes left to decay.

The spare gear should be in accordance with the Rules.

The emergency generator, if fitted, should be tested.

Remote stops for pumps and fans should also be tested at the Safety Equipment Survey.

Rotating Machinery

When repairing commutators the thickness of any replacement mica or micanite between segments is of the utmost importance and should be identical, otherwise the V ring will not seat properly, resulting in a breakdown of their insulation. Mr. Morris states under Repairs to Machines "Where windings are part renewed the new windings should be given a full high voltage test before connection to the existing windings". I am of the opinion the test should previously be extended to the remaining windings also, as one may be confronted with removal of the renewed parts to rectify a deficiency in the parts which were not renewed. The test should be held at the outset of the repair and of a voltage equal to that which will subsequently be applied to prove the new work. A satisfactory 500 V megger test is not an absolute guarantee of a satisfactory high voltage test. When engaged in high voltage testing I have megger tests held before and after for comparison. Would Mr. Morris inform me of the minimum insulation reading he would accept as a safe limit to proceed with a high voltage test, the reading obtained with a 500 V megger and the instance applied to machinery which has been standing for some time. Would Mr. Morris care to say a few words about the balancing of rotors and armatures and if it is practice to attend such tests and indicate the size at which we should become actively interested.

Has Mr. Morris any views in connection with spare gear for essential motors. I notice from the Rules that where motors engaged in essential service are likely to be duplicated then brushes and spanners are all that are required, but in dealing with steering gear where one motor or one motor-generator comprises the total equipment then an armature and field coils are required. Can Mr. Morris inform me why the windlass escapes attention. An instance occurred where a large tanker was delayed about nine days whilst a replacement windlass armature was obtained from the makers in Germany. Spare armatures were available on board for other essential motors. The local Port Authorities concerned refused to handle the ship without power on the windlass. Many owners do supply spare gear for the windlass as for the steering gear.

I was indeed enlightened to learn that automatic circuit breakers will require to be of an approved type, sometime in the future. This loophole was used recently when approved fuses and carriers were not available, an aged automatic circuit breaker was fitted.

I hope that some manufacturers of switchgear will be reminded of the necessity of fitting bolts in such a manner that they will not drop out if the nut slacks back. The Rules state that all nuts and screws which are used in connection with current-carrying parts and working parts are to be effectually locked so that they cannot become loose. The old method of nut and lock nut is being rapidly replaced by the nut and spring washer; I assume this is effectual. Does Mr. Morris share my view that a spring washer has a limited life?

The contribution which I have made is based on cases other Surveyors may also encounter in their outdoor duties and I have to thank Mr. Morris for the opportunity to pass them on.

Mr. A. C. Bailey

Mr. Morris is to be congratulated on his practical paper. It goes far to supplement the Instructions to Surveyors.

In his remarks on switchboards I feel he could have stressed the importance of positioning back-up fuses. They should be connected, ideally, to the bus-bars. Certainly not at the remote end of unstiffened or feeble connectors, thus inviting unprotected, internal faults.

Again, patent type cable terminals require the Surveyors' careful attention. Various sizes of "Klippon", as illustrated by the Author, mate with their respective cables based on the cable conductor's cross-section. For example a terminal might be suitable for a cable of, say, ·03 sq. in. This conductor size in general purpose rubber is rated at 60 amps; in butyl it is rated at 93 amps. It is wise then to check that all terminals are mechanically and electrically suitable.

Regarding the installation of cables, the following points could be mentioned:—

(1) Rigid P.V.C. conduit should be bent warm. Bent cold it does not take a permanent set. Subsequent failure of the clips may cause the conduit to spring straight with disastrous results to the wiring.

(2) P.V.C. sheathing prevents corrosion of armoured cables only when it is not slit by "drawing-in". Unfortunately, slits or punctures are difficult to observe.

(3) M.I.C.C. cables where possible should be arranged for top or side entry. Bottom entry to equipment, particularly in boiler rooms or deck-houses, may cause moisture to lodge on the seals with ultimate penetration.

Finally, battery compartment natural ventilation. The Dutch Ministry recommends, with capacities of 200 Ah or more, ventilation pipes of 6 in. diameter, with explosion relief valves on compartment doors. Recommendations are also made regarding the outlet position. On coasters they could be within "firing range" of cigarette smoking personnel. These recommendations may seem extreme. Nevertheless, ventilation arrangements are worth scrutiny.

Perhaps the Author may wish to record the German opinions in his reply.

Mr. E. A. PICKERING

I wish to thank the Author for the guidance given in his paper and for the information which we are sure to receive in the printed discussion. A paper such as this is extremely useful to the Engineer Surveyor stationed in a country which has no national marine electrical standards. The Author's comments on the following points would be appreciated.

Page 1 Factory Work

If a piece of equipment is ordered to L.R. requirements although non-essential it does not seem correct to accept National or I.E.C. standards in place of the Rules. The "Recommendations for Electrical Equipment", Section 4, does not seem to agree with the Author on this point. Perhaps the Author would confirm that paragraph M 410 would be included in the "recommendation" that non-essential machines be constructed according to Section M 4 of the Rules.

Could the Author state the principal differences between the I.E.C. standards, German standards and "the Rules" where they cover the same ground?

Page 2 Windings and Insulation

Many essential machines ordered to L.R. requirements are labelled as Class A insulated which I understand is normal for motors with mica to cores or frames, the remainder of the insulation being Class A.

It seems that the installation surveyor's only course is to accept the maker's statement in catalogues that the mica insulation is there for marine motors. Works test certificates sometimes have no indication of the insulating material used and whilst on the subject of these certificates many do not have sufficient information to check that they comply with Rule requirements without further enquiries being made.

Page 4 Self-Regulating Alternators

Has the Author any experience with and could he state the Society's attitude towards the self-regulating alternator marketed by Société Gramme, Pantin? I understand that these alternators may be paralleled without any attempt at synchronising and that short circuit protection in the switch gear is unnecessary as the excitation is automatically reduced under short circuit conditions.

Page 5 Voltage Regulation Tests

Occasionally a diesel generator set on installation fails to conform to voltage regulation requirements entailing considerable work on governor adjustments and engine examination. This invariably brings comment on the fact that the generator was tested and the engine was tested at the suppliers but the combination was not tested. If a generator set is ordered to our survey is the customer not entitled to a certificate for the set and not certificates for the individual items?

Page 9 Preference Trips

The Author states that a preference trip system operated by auxiliary contacts on the generator circuit breaker overloads is not now permitted by the Rules. Could he please point out in which way this is prohibited by the Rules?

Page 11 Labels

It does not seem to be correct to mark each fuse position with either cable size *or* fuse position and the fuse size. Not all ships carry personnel who would know what current capacity applies to a particular cable size and type. If the labels only state size of cable and fuse should a cable current rating list not be placed close to the switchboard?

Page 12 Cables

The words "clipped together" are used in mention of examination of cable installation where different types of cable are used. The Rules use the words "bunched together". Surely there is a difference between clipped together (possibly flat on a tray) and bunched together.

A list of recognised heat resisting natural and synthetic rubbers would be a useful addition to the paper. A list of those cables which the Author would consider flame-retardant and flame extending would also be useful. Does a P.V.C. sheath over lead make a lead sheathed rubber insulated cable flame-retardant? How does the Author interpret "bare lead sheathed cable"—bare externally or without proofed tape over the insulating rubber? Does the Author know why proofed tape over rubber required in the old Rules was omitted in the new Rule requirements?

The requirements of paragraph 1605(b) for cable support have puzzled me for some time as there seems to be less danger here than in the 'tween deck itself which may have large numbers of cables passing through it. I had the impression that some years ago all cables passing fore and aft across tanker bridge spaces were raised and passed around the house side whereas now it is common practice to pass cables through the bridge 'tween deck space. Although common practice it seems dangerous and contrary to the exact wording of paragraph 1604. Has the Author any comment on this?

How does the Author view the substitution of large cables by bus bars suitably taped and suspended and enclosed in sheet steel casing within the engine room?

Mr. H. G. DONALD

First, I must congratulate the Author on having compiled an up-to-date paper which, on account of its practical nature, is of interest to us all.

I find myself in complete agreement with the Author's notes, and this contribution is intended only to add something to the framework of the paper.

Windings and Insulation

Although the use of Class F and H insulating materials may be used only with special approval, I feel that this opportunity should not be allowed

to pass without some mention of the troubles that have been experienced. The use of silicone insulating materials in enclosed direct current motors and generators will produce abnormal brush wear. This fact has been known for a number of years. However, it may be of interest to some of the Surveyors who are not fully aware of the adverse effects of very small amounts of silicone in enclosed D.C. machines. This was recently noticed on the propulsion generators of submarines which had a very rapid brush wear immediately after being overhauled. In one instance, newly installed brushes wore down to the rivets in approximately three weeks operation, and the carbon dust resulting from the rapid brush wear caused a rapid deterioration of insulation to ground.

Extensive tests and investigation showed that the rapid brush wear was caused by the use of silicone rubber insulated wire for connecting the shunt fields, and silicone impregnated glass-mica-glass tape applied to some of the interior bus work.

After removal of the shunt connecting wires and the insulation from the bus work the armatures were cleaned and when the generators were returned to service, the brushes resumed their original very light wear rate.

The mechanics of silicone effect on brush wear is not completely understood. However, it appears that when even minute amounts of silicone vapour are absorbed in the carbon brushes, the silicone is converted by arcing under the brush face to a sand-like abrasive material. Another theory is that the water-resistant properties of silicone prevent the water vapour in the air from reaching the brush contact surface, which is necessary for forming the proper commutator film.

Whatever the reason, it is known that any amount of any type of silicone varnish, compound, rubber, grease, laminate or binder, will cause abnormal brush wear and should be avoided in enclosed direct current machines or enclosed A.C. machines having collector rings within the machine enclosure.

The foregoing should not be construed to mean that silicone materials are not suitable to be used on A.C. motors and generators, transformers, or other types of equipment that do not have commutators or sliprings within the enclosure. Also, silicone materials may be successfully used in parts of well-ventilated open D.C. motors and generators in which the incoming air passes over the commutator before entering the winding.

Location and Protection of Equipment

I agree with Mr. Morris that it is prudent to pay special attention to ventilating trunks, particularly at switchboards and generators, to ensure that there is no danger of sea water or rain running down them. However, there is another consideration which should not be overlooked and that is the positioning of the engine room supply fan intakes. In two sister vessels on which I was involved in the survey of the generators, all three of the generator armatures in each ship had to be completely rewound within a period of two years of the ships being new.

The engine room intakes of these vessels were located at the base of the funnel which was of the low streamlined type, and every time the soot blowers were used, the carbon from the boilers compounded with oil and dirt was deposited on the generators, thereby causing a rapid deterioration of the armature insulation which eventually broke down to ground.

After the armatures were rewound and filters fitted to the engine room intakes no further trouble was experienced.

Motors and Control Equipment

It may appear to be rather superfluous to mention that drip-proof equipment should be mounted vertically, but in a number of cases which have come to light recently, the equipment has been installed at 90° from the vertical.

Mr. R. I. MURCHISON

The Author hopes that his paper will be of assistance to his Engineer Surveyor colleagues.

I am sure it will, but more than that, it will be invaluable to the newly joined Electrical Engineer Surveyor in helping him to apply the Rules in a practical way until such time as he finds his own feet.

Even those who have served an apprenticeship at electrical engineering could not possibly have covered all the ground dealt with in this paper, and as their subsequent experience prior to joining the Society would likely be more specialised and correspondingly narrowed, they cannot bring much practical experience to their aid in interpreting the Rules.

When I joined the Society I would have welcomed a paper such as this to set me on the right road and avoid some of the painful pitfalls awaiting the new Surveyor.

Previous papers dealing with the survey of electrical equipment have been mainly based on United Kingdom practice and it is very refreshing to have a picture of the manner in which the requirements are implemented on the Continent.

In this respect it might not be considered outside the scope of the paper to have included some personal impressions on the way the "foreign" Surveyor is received by the different individuals he meets in the course of his duties.

I should be interested to know the extent of any language difficulties and how much the psychological approach has to be considered in the day-to-day dealings with the management and staff since, in the United Kingdom, there appears to be a difference in the atmosphere in carrying out a survey on a British ship as compared with a foreign ship.

I like the Author's sincerity in stating when he is not in accord with the Rules. I must reluctantly disagree with him, however, in one such instance dealing with fuses and rotary switches. I feel it is better for consistency to maintain the fuses on the outgoing side of the switch. One normally

expects the fuse to be "dead" when the switch is open, and any deviation from this arrangement, particularly on a switchboard, might result in injuries to personnel.

The Author in his paper covers a great deal of ground, and the overall picture tends of necessity to emphasise the major items of the Rules.

In my experience the major items tend to look after themselves and I find myself having to concentrate on the more insignificant items.

I would be prepared to state that the success or failure of a vessel's electrical installation largely depends on the attention given to the minor details.

May I give examples of just two such items which I find it necessary to follow up closely. They are earthing of fittings, and distance between steam pipes and cables at bulkheads of tankers.

In the first example it is not enough to see that there is a visible earth connection. It is well worth while making periodic check tests with a buzzer. It is surprising how often the visible earth connection has not been satisfactorily earthed. Furthermore, the sight of the Surveyor making such a test has a beneficial effect on the workman to encourage him to make a good job of the earth connections.

In the second example, the distance between steam pipes and bulkheads, the Rule requirements are quite simple, but it is surprising how often they need to be drawn to the attention of the interested parties and here we might vary the old adage by introducing an Irish flavour and say that prevention is much more pleasantly achieved than cure.

I would heartily endorse the Author's point on noting the information on machine nameplates, but not solely for its face value. It is very good discipline and the acrobatics that are often entailed in obtaining these particulars are amply repaid in the variety of information that can be picked up regarding the installation not all necessarily related to nameplate particulars.

With these remarks may I take this opportunity of congratulating the Author on his extremely informative paper which will be very useful in my opinion to anyone engaged in electrical surveying duties. May I say that it has drawn my attention to one or two items in the Rules on which I was not, shall we say, exactly up to date.

Mr. J. M. GARDINER

We now approach the Silver Jubilee of the Electrical Engineer Surveyors' previous "Vade Mecum" and it is heartening to see that, following the recent revision of the electrical section of the Rules and the issue of the Instructions to Surveyors, Part 6, a most praiseworthy effort has now been made, further to assist in the practical application of the requirements of Chapter M.

The following remarks are intended in the same spirit.

(I) (a) Rotating Machines (i) During Construction

The practice of quoting motor ratings in kW can be most misleading. To the writer's knowledge two instances have occurred in which installations were planned on this basis, the circuit currents, however, being evaluated on the assumption that the kW value was input to, instead of output from the motors. The underestimate of approximately one amp per kW resulted in undersized cables, switches and fuses.

The h.p. limit given in paragraph M 427 is mentioned, but can the Author state whether the values quoted in that paragraph or those quoted for electrical machines in paragraph G 103 should be used? Incidentally, the fee scale for the survey during construction of electrical machines commences at 100 kW.

Self-regulating Alternators

In connection with self-excited alternators it might be mentioned that if a machine has not been in service for some time it may be found, when next it is run, that the residual magnetism is insufficient to enable the excitation voltage to build up to the no load value. To overcome this disadvantage a small single phase autotransformer can be connected through suitable control gear to the alternator output terminals, and through a rectifier unit to the alternator field. This circuit, when energised, will provide the initial boost, then the normal no load excitation circuit will take over.

In cases where the self-exciting units are arranged for separate mounting, either in the switchboard or as free standing units, the witnessed test of the alternator should be run in conjunction with its respective equipment, not with the test-room equivalent.

(I) (b) Switchgear, Wiring and Connections

It is pointed out that for essential connections for propulsion equipment a certain minimum cross-sectional area is required, but it is equally important to emphasise that the wiring should be stranded—having not less than seven strands.

The Author's opinion concerning the use of compression type cable shoes is endorsed, with perhaps a reservation in the case of small flexible connections to the moving contact of contactors. This qualification is prompted because of a recent experience of a steering gear failure due to such a connection snapping inside the bell mouth portion of the cable shoe, just where it was compressed. As this equipment, for the operation of the telemotors, was of German manufacture it would be interesting to learn if the Author has had similar experiences with such connections.

Circuit Breakers

In Sweden the arrangement of main circuit breakers follows the Continental pattern. However, for the feeder switches and fuses a new method is being developed. A switch and fuses are mounted on an insulated base and the captive

bolts connecting the terminals to the busbars extend through this base. Those bolts may be tightened or unscrewed from the front of the board by means of a special insulated hex-headed driver. Once the base fixing screws have been removed the complete unit may be withdrawn for servicing, a compromise between British and Continental practice.

The location of the reverse current trips for D.C. generator breakers is specially mentioned and it is considered that this would be an appropriate opportunity to draw attention to the ammeters for such generators. The ammeter should be connected in the same pole as the reverse current trip, that is, in the pole opposite to the equaliser connection to the series field. It follows from the requirement of M 420 that this should be the positive pole. However, on all occasions when the number of parallel running generators on a vessel is to be increased, the polarity of the series field should be checked on the existing and new machines to make sure it is the same in both cases.

EARTHING

Another system for the detection of earth faults on A.C. installations consists of a transformer rectifier unit the output of which can be switched to inject a D.C. voltage into the various networks, the condition of the installation being indicated by a voltmeter, calibrated in ohms.

(II) NEW CONSTRUCTION SURVEYS (vi)

It is noted that the switchboard certificate required in M 619 has not been included in the list of certificates requested. If this omission is intentional, the purpose of requiring the switchboard manufacturer to furnish a certificate is not clear.

Cables

(a) DURING INSTALLATION

The method of separating the two runs for the duplicate feeder cables to the steering gear motors on cargo ships has been stated. It would be helpful if the arrangement in the case of oil tankers could also be given.

CABLE SUPPORTS

The practice of laying cables in troughs of perforated metal can be satisfactory if the degree of bunching is limited and if clips are arranged so as to restrict possible movement of the cables.

CABLES ON TANKER GANGWAYS

It should be emphasised that where cables are run in pipes along gangways care should be taken in the arrangement of cable expansions in way of pipe expansions to ensure free movement without chafing of the cables.

CABLES THROUGH BULKHEADS

The use of boxes filled with an epoxy resin compound as fireproof bulkhead glands is of interest. It was found after a standard one hour fire test conducted on such a trunk that the minimum length of trunk necessary to satisfy the test regulations was approximately 6 in. (150 mm.) on each side of the bulkhead. Following this test the gland was tested for degree of watertightness. With a head of water of 20 ft. (6 m.) the leakage was approximately two litres, say half a gallon, per minute—mainly through the cables.

LOCATION AND PROTECTION OF EQUIPMENT

With the introduction of "dead front" type switchboards a feeling of false security may appear. On open type switchboards a fine sloping canopy was fitted to protect the equipment from drops of condensation or other accidental "wettings". Now with the boxed-in boards one is apt to assume that all is well, but an examination of the flat, non-watertight covering on top of the cubicles, usually reveals many places requiring attention—or the provision of a fine sloping canopy.

EARTHING

It is important to ensure the earth continuity of the lead sheaths of sub-circuit cables at all terminations—lighting fittings, switches, junction boxes, etc.—especially so with textile braided or P.V.C. sheathed lead-covered cables. Otherwise, only the two supply cables for each sub-circuit will be earthed at the distribution board.

CABLE TERMINATIONS

It is fairly common on A.C. installations where large currents are involved, to restrict the sizes of three-core cables to reasonable handling limits and obtain the necessary cross sectional area by connecting a number of three-core cables in parallel. The following may appear elementary, but it is surprising how frequently the fault occurs, mainly in cases where there are three or six cables in parallel. The cores from each three-core cable should be connected to their respective phase terminals on switchboard, alternator, etc., and not connected together to form a single-core cable.

MOTORS

Some types of windlass are fitted with an electric motor of the flange mounting type for bolting to the windlass casing. The terminal box is mounted on the side of the motor and the cables are usually led to this through a straight pipe extending from deck to terminal box. In such a case it is advisable to arrange an expansion joint in the pipe as otherwise, due to the vibration when the windlass is working, there is every possibility that the terminal box will be smashed or its securing bolts sheared.

(b) OVERLOADS

Steering gear motor control gear, although not fitted with overload protection, still requires overload relays to operate the alarm stipulated in M 616. These relays should be fitted in at least two of the three phases for A.C. systems. If the owner so desires, overload alarms instead of over-

load protection may be used also for essential motors which are duplicated.

(c) No-voltage Releases

On certain American-owned oil tankers the motor control gear for the main lubricating oil pumps; control air compressor; service air compressor; sanitary pumps; drain pump; steering gear motors, main and auxiliary condensate pumps, is so arranged that in the event of a power failure the contactors open, but these units will automatically restart on restoration of the supply.

Dangerous Spaces

One small item concerning lighting fittings in pump rooms appears in M 1608 and the intention may not be entirely clear. "The lamps and corresponding switches . . . are to be suitably labelled for identification purposes." The purpose of the label at the lighting fitting is to make certain that anyone intending to replace the lamp, or do other work on the fitting is aware of the location of the controlling switch.

Battery Room

This is one compartment which deserves the closest of inspection to ensure that no equipment is installed there which could cause the ignition of explosive vapour, nor any chance of the vapour finding its way through unpacked cable glands, deck tubes or other openings into a space which normally would be considered safe.

(III) Periodic Surveys. Electro-magnetic Slip Couplings

One item of this equipment which requires special attention is the brushgear and sliprings for the field coils. The insulation between the rings should be kept absolutely clean as the voltage induced on the opening of the field circuit—even although there is a discharge circuit—will not be long in tracking across an oily carbon film.

AUTHOR'S REPLY

The Author is indebted to all those who have taken part in the discussion as well as to those who have made written contributions.

The diversity of the comments made and questions asked contain a wealth of information and experience gathered from all parts of the world and which will not be found in text books. As such they make a valuable supplement to the paper.

It is hoped that the following replies will also make a suitable contribution.

To Mr. Haffner

Commutators

In Germany this type of commutator is generally used for small machines, although I have seen some up to 12 in. in diameter, made in this way.

The movement of the parts in the commutator concerned occurred in service. This fault could not have been spotted during survey, and that is one of the reasons why I treat them with suspicion.

Temperature Tests

It is intended that the thermometer should read the highest temperature.

Switchgear

The mechanical strength of insulating materials is mentioned in the last paragraph of page 6, and I fully agree that this is most important.

Circuit Breakers

The statement regarding equal interrupting capacity is incorrect.

The breaking capacity of each interrupting device must be large enough to deal with the prospective short circuit current expected at the point of installation. Thus the generator circuit breaker or fuses need not have such a large breaking capacity as an outgoing feeder's circuit breaker or fuses.

Preference Tripping

I agree with Mr. Haffner's remarks relating to the switches cutting out the preference tripping circuits. These are, of course, intended for use in port only, as during this time the non-essential drives such as cargo winches become the most important auxiliaries on the ship, since they earn the money for the owner. These drives are invariably connected to the preference circuits, and tripping during loading of cargo causes delay.

I am glad to learn that time-based settings are acceptable for preference tripping circuits, although I must admit that I had interpreted M 609 to mean that this was not in order. As the generator overload trips are quite often set at 150 per cent as recommended in the Rules, this interpretation seemed reasonable to me, bearing in mind the action of a diesel-generator set on overload. Furthermore, of course, current settings are given for these circuits in the Recommendations to the Rules.

Cable Supports

I have never seen cable troughs which have been tailor made for the cables, and in my opinion the cables should always be clipped to prevent movement, even if the clips are two to three yards apart.

Galvanised steel screws or bolts are quite suitable for securing the metal trays although aluminium bolts can of course be used provided the heads are insulated from the tray plate. Wooden grounds are also often used but care should be exercised in selecting the wood.

Switchboards

There never seems to be any room for the switchboards. When fitting out, it seems to me that the switchboard position is made the centre of gravity for all the pipe flanges and valves, not forgetting the tank sounding pipes.

I have found that the only answer to this problem is to insist, even with "dead front" boards, on having a properly constructed weather-proof canopy fitted over the switchboard. The canopy should of course be provided with a drain-off.

Dangerous Spaces

This is a very difficult problem, and by the time the Electrical Engineer Surveyor has seen this—even on his first visit after launching—it is generally too late. By this is meant that an alteration, either electrical or constructional, has to be undertaken.

I am convinced that the only satisfactory answer to the problem is for the electrical plans to be considered in conjunction with the structural plans.

Discrimination

A switch and fuse is always selected by the Author due to the possibility of injury to personnel and/or damage to the circuit breaker.

To Mr. Mort

Fixed type circuit breakers are preferred because they are cheaper and can be permanently secured. They are, therefore, not so liable to move and cause mal-operation in service.

Such mal-operation, due perhaps to vibration, can be caused by poor contact in the auxiliary holding or tripping circuits, particularly if these contacts are spring-loaded and not plugged.

It is also considered that if repairs have to be carried out, then the switchboard should be made dead, as the ship's movement makes it, in any case, a dangerous practice to work on a live board. It is difficult to imagine the sort of repair that *must* be carried out at sea (i.e. which cannot wait until the vessel is in port) and which would not involve shutting down the complete switchboard.

To Mr. Fennell

1. I agree that mica should be used.
2. Many manufacturers are not prepared to allow the Surveyors to be present at such tests.
7. As there is no suitable flame-proof fitting as yet available for hydrogen gasses, a robust watertight/gastight lighting fitting is permitted in these compartments.
8. My remarks regarding earthing of cabin fittings were meant to imply that *each* cabin was provided with its own earthing system. Incidentally, some contractors prefer to run three-core cables to the fittings and use the third core as an earth, which is then made in the alleyway.
10. It is normal international practice to omit separate pilot lamp fuses in control gear up to 25 amperes (British practice as per the I.E.E. Regulations is 15 amperes) except where failure of the lamp would jeopardise the supply to a piece of essential equipment. The fuses, of course, provide only short-circuit protection.

To Mr. Tickell

Winding and Insulation

Polyester based enamels are often used for the windings on rotating parts of Class E machines and are permitted by the Rules. To the best of my knowledge they have given no trouble.

Self-Regulating Alternators

Generally speaking the silicone rectifiers produced in Germany are, as far as I am aware, giving satisfactory service. In some countries, it is the practice to grade the rectifiers in accordance with the break-down voltage, and there are often three grades. This is easily understood when the size and consequently the creepage distances of such rectifiers and also the heavy costs of manufacturing them are considered. The only trouble I have experienced with them has been due to breakdowns through voltage surges, which have been caused by carrying out tests on the alternator and where in my opinion the rectifiers were not of the highest grade.

Transformers

Where possible liquid transformers should be avoided, but if there is no alternative, then I prefer that the liquid should be non-flammable and non-toxic. There are several of these synthetic insulating oils, and special precautions for handling them are generally necessary. The windings should also have special insulating materials. Such a transformer should be fitted with a conservator, oil level indicator and silica-gel breather, and the design should also take into consideration the rolling and pitching of the ship.

The Buchholz relay is a gas operated device connected between the transformer tank and the conservator. As any fault which occurs inside a transformer is generally accompanied by the evolution of gas, such a relay should be fitted and provided with alarm and tripping contacts to give the necessary protection.

It should, however, not be forgotten that a new transformer gives off bubbles of air and therefore it is usual to defer connecting the relay tripping contacts for a short period.

Transistor Control

The alternative control circuit should, in my opinion, be another sytem altogether.

The draw-back to transistor control equipment being used for such an essential auxiliary as the steering gear, is that it cannot be repaired and it is not possible to check the condition of the component parts, as these are generally sealed in compound.

However, the reliability of such equipment is continually improving and transistors are being used with increasing frequency on board for alarm circuits.

DANGEROUS SPACES

It is regretted that the comparison required by Mr. Tickell between the VDE Standards 0170/0171 and the BSS.229 is outside the scope of this paper. Such a large and complicated subject is best left for a separate paper.

TO MR. CRUM

Transient voltage response tests on self-regulating alternators are not routine works tests, but type tests already carried out by the manufacturer.

My initial reaction to the static excitation equipment was the same as Mr. Crum's, namely, that a hand regulator should be fitted as an alternative. However, a very large proportion of the new buildings that I have surveyed in Germany since 1956 have been fitted with self-regulating alternators without hand regulation and, apart from the case mentioned, in replying to Mr. Tickell I have not heard of one complaint of the regulating devices being out of service.

As indicated in my reply to Mr. Tickell, the answer probably lies in fitting better class rectifiers, as these, from my own experience, are generally the weak links.

The opinion of the German manufacturers, backed up by experience, and one which I agree with, is that adjusting devices such as trimming switches, are a source of potential trouble due to them being played with by the engine room staff particularly when carrying out paralleling operations. For this reason, they are either not fitted at all or are concealed behind the generator switch panel doors.

It is assumed that the "high torque" type squirrel cage motor mentioned by Mr. Crum is the double squirrel cage motor, and these are fitted in preference to the normal squirrel cage motor. Their starting currents are, however, generally of the order of five times full-load current, which under certain circumstances can be critical.

The answer, of course, as Mr. Crum suggests, is to discuss the matter with the electrical drawing office.

TO MR. WRIGHT

FACTORY WORK

The description of the factory work covers the survey normally carried out by a Surveyor visiting a new factory or when taking over a factory in a new district. The items detailed are not often examined in one visit, but are generally spread over several visits.

Trouble with the insulation of internal leads in machines is not uncommon and can generally be traced back to the practices used in the factory.

The use of P.V.C., P.C.P. or rubber insulated coil end flexibles is made possible by the different methods of motor manufacture. For instance, one practice is to fit suitable lugs to the ends of the winding wires to provide connections for the flexibles, which are attached *after* impregnation and assembly of the windings. Because such coil end flexibles have to withstand only the conditions existing in the air space between the windings and the terminal block, it is probable that a Class B machine could be safely wired in this way with a relatively low temperature flexible.

Another practice, however, is to attach the flexibles to the ends of the windings at the assembly stage. Flexibles so attached are actually built-in so that the connecting ends are therefore subject to the full temperature of the machine windings. Furthermore, when the complete assembly is stoved during the impregnating cycle, the coil end flexibles reach full oven temperature long before the main mass of associated windings. The insulation of such flexibles must, therefore, be designed to withstand not only the stoving cycles, but the winding temperature at the end embedded within the windings.

The flexibles must also withstand any penetration or solvent attack arising from the impregnating varnish during the dipping operation on the complete assembly.

When it is remembered that the operating temperatures of Class B machines can be as high as 130° C., not forgetting that the impregnating temperatures during manufacture can be even higher, it will be appreciated that neither P.V.C., P.C.P. nor rubber insulation is ideally suitable for these connections. The best solution is to use glass or a similar insulated flexible.

PERIODIC SURVEYS

I fully endorse Mr. Wright's remarks regarding discussing the installation with either the Chief Engineer or the ship's electrician and also that the recommendations should be given to the owners' representative.

TO MR. COOK

PERIODIC SURVEYS

I agree with Mr. Cook's remarks, but it should be pointed out that the examination of the spare gear is no longer a survey requirement.

ROTATING MACHINERY

I would never recommend a high voltage test to be applied to existing windings, as this could damage the insulation without breaking them down, thus leaving the windings in a worse condition. On the same basis, even with new equipment, high voltage tests should be carried out very sparingly and the full recommended voltage applied only once when new. Thereafter, the maximum value should not be more than about 75–80 per cent of the full value and this also on new, clean windings.

There are no figures given for the minimum values of insulation resistance, but if the Rules

or Standards call for a high voltage test to be carried out, then—unless this is done—the equipment is not acceptable.

With regard to the balancing of rotors and armatures, whether these tests are witnessed or not should depend on the Surveyor concerned and his opinion of the factory or workshop.

The Rules do not call for a spare armature for the windlass motor, and I would not presume to comment on this, except to say that the windlass duty cannot possibly be compared with the steering gear duty. In an emergency, some means would always be found of either raising the anchor or disconnecting it from the ship.

I agree that spring washers have a limited life, but they can of course be very easily replaced.

To Mr. Bailey

I was rather surprised to read Mr. Bailey's remarks on M.I.C.C. cable seals, as these are not borne out by my own experience. The seals are compound filled and the cores sleeved with insulating material, generally neoprene, so if the seals have been penetrated, then they have not in my opinion been made properly.

There are no German statutory requirements for the battery compartment ventilation, but the Dutch Ministry recommendations indicate the importance of adequate ventilation in such compartments.

To Mr. Pickering

Factory Work

The "Recommendations for Electrical Equipment" are not mandatory, so the Rules do permit non-essential equipment to be manufactured to National Standards or I.E.C. recommendations. In actual fact, of course, these latter Standards, in practically all important aspects, are generally equivalent to the present Rules.

My interpretation of the recommendation referred to by Mr. Pickering would not require the additional insulation specified in paragraph M 410 to be fitted. After all, the machine is "non-essential".

Windings and Insulation

If a doubt really exists that the additional insulation required by paragraph M 410 has been fitted, then I can only recommend that the machine be refused until the manufacturers can prove their statements to the Surveyors' satisfaction. Is it not possible to visit the factory, or, if the factory is not local, to take the matter up with the Surveyors at the appropriate port?

Test certificates not containing sufficient information should be returned to the shipbuilders at once, advising them of this fact and pointing out that the Rules call for fully documented test certificates.

This trouble was initially experienced in Germany, but by being ruthless in returning the test certificates, the manufacturers quickly co-operated rather than again risk the anger (and custom) of the shipbuilder.

Self-regulating Alternator

I am afraid that I have no experience of the alternators referred to, although I would imagine that discrimination in the operation of the protective devices would be rather difficult to achieve in such a system.

Voltage Regulation Tests

This trouble could be cured if the engine manufacturer specified the governor regulation to the generator manufacturer. He has to do this now in any case for all diesel alternator sets intended for operating together in parallel.

Preference Trips

This is dealt with in my reply to Mr. Haffner.

Labels

The Rule calls for the current carrying capacity of the circuit and the fuse size to be given on the label. As stated in my paper, it is generally more useful for the ship's personnel to know the cable size when it comes to replacing the cable. However, if the cable size is given, then the insulation should also be stated.

Cables

I agree that there is a difference between bunching and clipping. To me, bunching has always meant that the cables have been tied together whereas clipping meant supporting and fixing them to a flat surface. Clipping might perhaps be said to be a particular method of bunching. As far as cable rating is concerned, bunching is the more arduous condition.

The insulating materials are graded in accordance with the temperatures they can withstand and as far as I am aware there is no list available other than that given in Table 8.1.

All unsheathed cables are flame-extending and generally speaking so are bare lead-sheathed cables hence the restrictions given in paragraph M 812. With multi-core bare lead-sheathed cables having unvulcanised regenerative rubber filler, however, it is interesting to note that tests to M 852 have proved the cables to be flame-retardant. This was due to the gas from the filler extinguishing the flame from the cable immediately the bunsen burner was removed.

From tests I have witnessed, a P.V.C. sheath over the lead will make the cable at least flame-retardant and possibly fire resisting as defined by M 850. However, this type test should be carried out by the cable manufacturer in each case.

A bare lead-sheathed cable has no further external covering or sheath.

The proofed tape was required so as to make sure that the conductor insulation was watertight. However, this requirement was relaxed when the rubber was applied by means of extrusion in association with continuous vulcanising as this process more or less guaranteed that the cables were watertight.

I agree with Mr. Pickering regarding paragraph M 1605 (b) and I believe that paragraph M 1604

is being amended to make it clear that through cable runs are permitted in the 'tween deck spaces.

Bus bars fitted in steel casings are excellent in my opinion provided that: —

(1) They are securely supported on insulators.

(2) The casings are mechanically strong enough to withstand rough handling without the panels being bent or pushed on to the busbars.

(3) The casings are watertight and are large enough to dissipate the heat.

(4) The casings are painted with a distinguishing colour, preferably red, marked up to show the voltage of the busbars.

To Mr. Donald

I was particularly interested to read Mr. Donald's contribution, and although I have heard of the problem of silicone "gassing", I have never had any personal experience of this.

To Mr. Murchison

Mr. Murchison's remarks regarding the reception given to a "foreign" Surveyor hardly fall within the scope of a technical paper, but it is my opinion that, as in the United Kingdom, the reception given to a Surveyor depends on the impression he has personally created on the various people he meets. Thus, in a foreign country, any attempt to speak the language, immediately creates a good impression and provides a fund of goodwill, which the Surveyor may be glad to draw upon later.

I would add that for an Electrical Engineer Surveyor overseas, who is concerned with actual every-day survey work and has a large district to cover, the language of the country is important and should be learned, otherwise it becomes extremely difficult to keep abreast of any technical developments, not to mention the disadvantages of not being able to speak with the electricians on the job.

Mr. Murchison's remarks regarding the location of the fuses are noted, but the "consistency" referred to by him does not apply to all ships and one should not expect the fuses to be on the "dead" side of the switch. From the comments made on this paper by other Surveyors, Mr. Murchison will see that there is a definite body of opinion in favour of the arrangement of fuse-switch, particularly when the switch is of the rotary type.

I heartily endorse his remarks regarding earth-testing and would recommend that this be regularly done, as it is very often the weak link in the chain.

May I add that I was once told by a very distinguished Engineer Surveyor that a good Surveyor was one who paid attention to details and that major items also had minor details.

To Mr. Gardiner

I have studied Mr. Gardiner's remarks very carefully.

(I) (a) (i) Rotating Machinery

It is surprising that such a mistake would be made by a designer, as motors are always rated by their output.

In my opinion, the values in paragraph M 427 should be taken, as after all, Chapter M contains the electrical Rules.

(I) (b) Switchgear

I would thank Mr. Gardiner for pointing out that the essential propulsion cables should be stranded as required by the Rules.

I have not experienced the trouble with compression sockets quoted by Mr. Gardiner. I have, however, heard from some manufacturers that, contrary to expectation, this trouble has been experienced with the smaller control cables having stranded cores and that they have gone over to solid cores because of this.

(II) New Construction Surveys (vi)

The omission of the switchboard certificate was accidental.

(a) Cables

The duplicate steering gear feeder and control cables on tankers should also of course be kept apart as far as possible, and it is good practice for one to be taken direct to the steering gear flat by the shortest route, whilst the other is run via the accommodation. Alternatively they can be run on the port and starboard sides of the engine room.

Cables through Bulkheads

It is interesting to note Mr. Gardiner's remarks regarding fire bulkheads. The Ministry of Transport requires that the length of the bulkhead box should be at least 15 in. and this, of course, on the insulated side of the bulkhead.

Switchboards

A weatherproof canopy with drain-off should always be fitted.

Motors

I was surprised to see the remarks regarding windlass motors, as many such arrangements have been built in Germany without the damage to the cable pipes mentioned.

No-voltage Releases

The omission of these trips on essential motors is often requested by Danish owners.

Dangerous Spaces

The German VDE Standards recognise this possibility, hence their insistence on a built-in interlocked switch in the E *x* (d) lighting fittings.

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THE NEW APPROACH TO LONGITUDINAL STRENGTH

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The New Approach to Longitudinal Strength

By J. M. Murray, M.B.E.

In the last ten years there has been a profound change in the approach to the problem of determining the forces and thence the stresses to which the main longitudinal girder of a ship is subjected. Formerly, the only way to derive the appropriate section modulus of a new design was to base it on experience of previous ships, by means of the classical static calculation. On the whole, this method has given remarkably good results, but the very strong element of empiricism in the comparison invites criticism. The new methods about to be described are intended to give a solution which is more theoretically correct than the present one.

In the history of the development of the main strength members of the steel ship the most remarkable feature is the continuity; in spite of the great increases in size and speed which have taken place in the last 100 years, no radical change has been made in the basic standard of strength. When the strength of the *Great Eastern* is compared with that of the *Queen Mary* it will be found that the difference is not great, and this continuity applies to cargo ships also. From this we may conclude that the standard set by our ancestors was reasonable, and both good judgment and luck may have contributed to this, and that the methods of extrapolating from these beginnings were sufficiently sound; otherwise, the hard experience of meeting the incalculable forces of the sea would have revealed the deficiencies.

For the greater part of this time extrapolation from previous experience has been made by means of the artifice always associated with John's classic R.I.N.A. paper of 1874. There was, of course, a strong background to John's paper dating from the work of Rankine. In time, refinements to the method of computing the strength of a ship by determining the bending moment when the ship was statically poised on a trochoidal wave were introduced. The Smith correction takes account of the distribution of pressure in the wave inherent in the trochoidal wave theory and the Read correction is intended to cover the effects of heaving and pitching on the hydrostatic forces. Considerable attention, too, has been given to the proportions of the wave used in the static calculation which John assumed to have a height of 1/28L. Later on a height of 1/20L was used and this became standard practice until about ten years ago a wave height $1{\cdot}1\sqrt{L}$ was adopted by L.R. to give a more realistic comparison between ships of different lengths than emerged from the use of a wave of constant proportions.

A few records of actual stresses obtained at sea serve to indicate that stresses obtained by means of the static calculation do in fact give a measure of the actual stresses sustained at sea. Measurements taken on the *San Francisco* and to a much greater extent on the *Ocean Vulcan* have shown that this was so for a certain class of cargo ship, and further records of stresses taken at sea have confirmed and extended this conclusion.

Nevertheless, it was always recognised that there was a strong element of artificiality in the basic assumption that the trochoidal wave gave a realistic representation of the kind of wave commonly met at sea. The sea in a storm is characterised not by its regularity but by its confusion and the regular wave is rarely encountered in such conditions, though it may be seen in the swell which follows the storm. The sea in a storm is composed not of regular long crested waves but of irregular short crested waves, and this results from the superposition of a large number of waves of different lengths and heights travelling in different directions. In fact, the storm sea can be represented diagrammatically as the sum of a series of corrugated surfaces, as shown in Fig. 1 which has been reproduced on many occasions. Each component can be regarded as a sine curve and a simplified example of the final effect can be seen from Fig. 2, in which three waves of different heights and periods are combined. It is on this basis that the new approach to strength is founded.

In what follows only the barest outlines of the theories are given, and many steps are omitted, some because they are obvious, but many more because the mathematics are very specialised and are beyond the capability of most naval architects. The reader must therefore be prepared to accept rather uncritically much of what follows, but he may derive some comfort from a remark by Professor Lewis that "some of the simpler concepts, at least, can be applied without a detailed understanding of the unusual methods involved".

The Sea

The fundamental principle involved in the new approach to the strength problem is that the motions of a ship in a seaway—and the bending moment comes in this category—can be regarded with sufficient exactitude as the linear superposition of the individual effects of the simple sine components of the seaway. It is therefore necessary to consider how the seaway can be expressed in a form which will allow this principle to be utilized. In a simple deep sea wave the following definitions and relations hold:—

$$\text{Wave length} = \lambda$$

$$\text{Period} = T = \sqrt{\frac{2\pi\lambda}{g}}$$

$$\text{Amplitude} = a$$

$$\text{Speed} = c = \frac{gT}{2\pi}$$

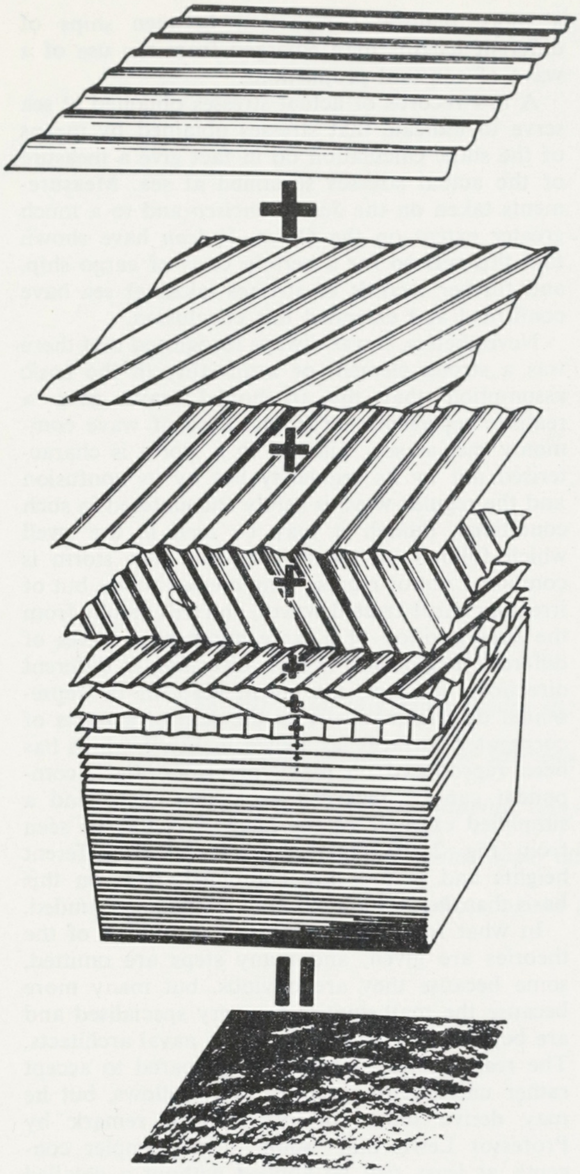

FIG. 1

$$\text{Circular frequency} = \omega = \frac{2\pi}{T} = \sqrt{\frac{2\pi g}{\lambda}}$$

In a simple sine wave the profile is represented by $f(x, t) = a \cos\left(\frac{2\pi x}{\lambda} - \frac{2\pi t}{T}\right)$

For the wave form given in Fig. 2 the resulting profile is given by equation

$$f(x, t) = a_1 \cos\left(\frac{2\pi x}{\lambda_1} - \frac{2\pi t}{T_1}\right) + a_2 \cos\left(\frac{2\pi x}{\lambda_2} - \frac{2\pi t}{T_2}\right) + a_3 \cos\left(\frac{2\pi x}{\lambda_3} - \frac{2\pi t}{T_3}\right)$$

and the figure shows the situation at intervals of three seconds.

The change in the form of the sea with time is continuous, and this simple combination of three sine waves has evidently some analogy to an actual sea shown in Fig. 9 (b). In this exposition

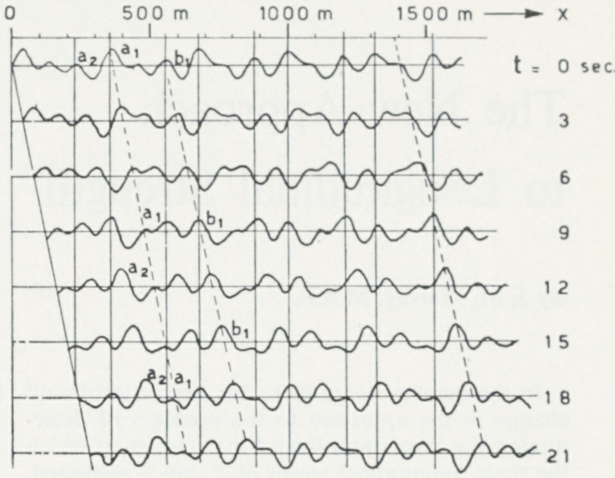

FIG. 2

Wave form, obtained by superposition of three simple waves

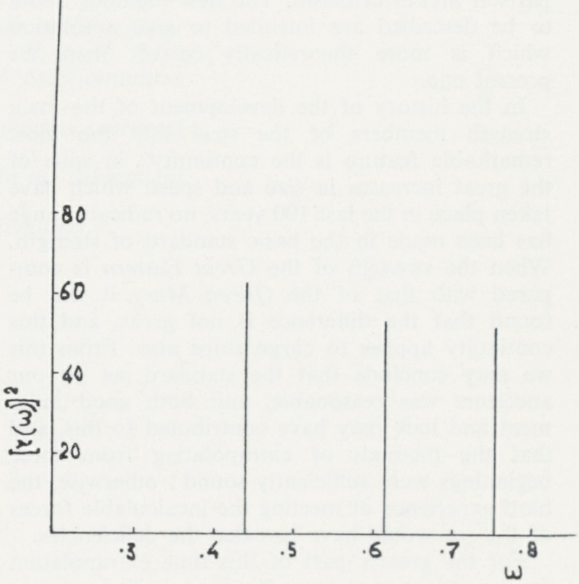

FIG. 3

Spectrum of waves in Fig. 2

the three dimensional aspect of the sea is sidestepped, but since important components of storm waves all travel in roughly the same direction, this simplification can be accepted. Evidently, by increasing the number of sine components of the wave the irregularity will increase and the actual sea will be more closely represented. Nevertheless, a combination of periodic functions will never completely describe the profile of the sea, for if carried on long enough the profile will repeat itself and the actual sea observed at any moment will never be repeated. A description of the wave at a point which will reflect the complexity of the sea and at the same time be realistic and readily

handled is required, and the representation which has been adopted by St. Denis & Pierson seems to be the only really suitable one. It is described in the next section, and has as its basis the superposition of harmonic waves at random phase,

$$r\,(t) = \sum_{n=1}^{n=n} a_n \cos(\omega_n t + \varepsilon) \text{ where } a \text{ is the amplitude.}$$

The Energy Spectrum

According to this representation the wave state at a fixed point is given by

$$f(t) = \int_0^\infty \sqrt{\left[r\,(\omega)\right]^2 d\omega} \quad \cos\left[\omega t + e\,(\omega)\right]$$

This is not an integral which can be formally integrated, but is merely the mathematical representation of a process; the random superposition of an infinite number of sine waves $\sqrt{[r(\omega)]^2 d\omega}$ is the amplitude of the component sine waves. The representation of $[r(\omega)]^2$ as ordinate on a basis of ω is called the energy spectrum. It has the dimensions of a squared length $\times$ time and must not be confused with f (t), which represents the wave record as a function of time. It measures the average squared value of the amplitude of the component waves associated with a particular frequency. The term "energy spectrum" is used because of the connection between $[r(\omega)]^2$ and the average energy of waves on the surface of the sea.

The cosine factor provides the oscillating character of the component waves. e is a random phase angle which can only be defined by saying that it can take any value between zero and 2π with equal probability.

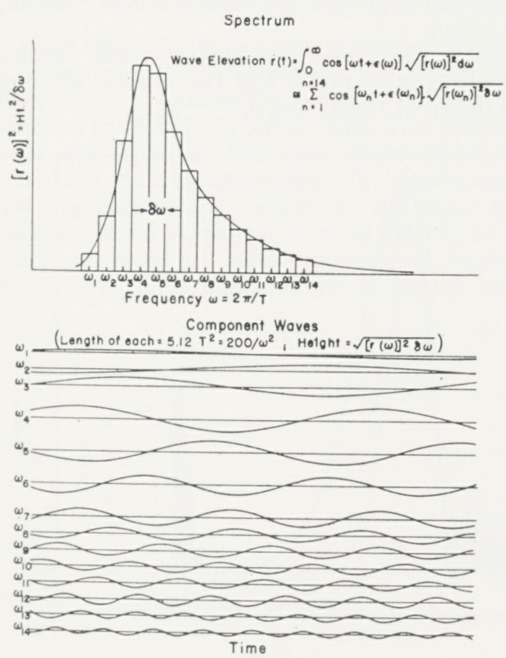

FIG. 4

Typical energy spectrum—showing approximation by a finite sum of components

In this form the equation represents any sea surface. For a particular energy spectrum this equation does not represent one sea surface but a whole series of sea surfaces which are all equivalent in terms of probability.

It is now necessary to explain rather more fully the physical meaning of the energy spectrum. Fundamentally, it is the sum of the energies of the component sine waves of the storm sea and it indicates how the energy contained in the waves is distributed over the component sine waves. The energy contained in the component waves is proportional to $\frac{1}{2} a_n^2$ where a_n is the wave amplitude. The storm sea is composed of an infinite number of harmonic components, and the energy spectrum is represented as a continuous curve with each ordinate $[r(\omega)]^2$ having a height of Fn (ω) on a basis of ω where Fn (ω) $d\omega$ is the energy of all components in the interval of frequency from ω to $\omega + d\omega$

$$\text{i.e., Fn}\,(\omega)\, d\omega = \sum_{\omega}^{\omega + d\omega} a_n^2 = \left[\, r(\omega) \,\right]^2 d\omega$$

(*see* Fig. 4).

Several important riders follow from this representation of the energy of the sea.

(1) The spectrum has two important properties—the area of the spectrum $2m_0$ and the relative width of the spectrum ε

$$2m_0 = \int_0^\infty Fn(\omega) d\omega \qquad \varepsilon = \sqrt{\frac{m_0\ m_4 - m_2^2}{m_0\ m_4}}$$

where m_2 and m_4 are the second and fourth moments of m_0 about the origin. When ε is small the energy spectrum is composed of a narrow band of frequencies and the swell condition predominates. When ε approaches 1 then the sea is very confused.

(2) $m_0^{\frac{1}{2}}$ = root mean square of f(t), i.e., the standard deviation of the heights of the wave from trough to crest about a mean value taken from a record of the wave profile which the sea spectrum in question represents.

(3) The statistical distribution of the maximum height h of the waves of which the spectrum is composed depends on m_0 and ε. When ε is small, i.e., when the wave state approaches the swell condition, the distribution of the maxima of h tends to a Rayleigh distribution, i.e., the probability that h is greater than a

$$p\left[a < h\right] = \frac{1}{m_0} \int_a^\infty h\, e^{-h^2/2m_0}\, dh$$

When ε is large and approaches 1, the distribution tends to a Gaussian distribution, i.e.,

$$p\left[a < h\right] = \frac{1}{(2\pi m_0)^{\frac{1}{2}}} \int_a^\infty e^{-h^2/2m_0}\, dh$$

It is generally more useful to know the distribution of H, i.e., the height from crest to trough, and if the spectrum is narrow then

this distribution is Rayleighan. The area of the energy spectrum $2m_0$ is usually described in terms of E where $E = 2\ m_0$.

(4) Furthermore, it has been shown that when ε is less than about ·5 the average heights H of the waves in the spectrum can be described by $1{\cdot}77\sqrt{E}$ or $2{\cdot}5\sqrt{m_0}$. The significant height, that is, the average height of the highest third of the waves $H_{\frac{1}{3}} = 2{\cdot}83\sqrt{E}$ or $4\sqrt{m_0}$ and the average height of the highest tenth of the waves $H_{\frac{1}{10}} = 3{\cdot}6\sqrt{E}$ or $5{\cdot}1\sqrt{m_0}$. Thus, knowing the area of the spectrum E the probable heights of the maxima can be deduced, but it should be remembered that these averages vary with ε. In fact, these relations are only valid when ε is less than ·5.

Energy spectra can be computed in several ways. In the first place a wave record is necessary and this can be obtained by pressure gauges on the sea bed, by a ship-borne recorder, by a wave measuring buoy or by analysing stereoscopic photographic records of the surface of the sea. From records thus obtained the spectrum may be obtained by expressing the wave record as a series of numerical values of wave heights at particular moments and then calculating the spectrum with the help of a digital computer, or the wave record may be obtained in a continuous black and white form and the spectrum obtained graphically by means of an analogue computer. A normal Fourier analysis is of little use in this problem as the first 20 or so harmonics can be neglected; it is the next 150 or so that count.

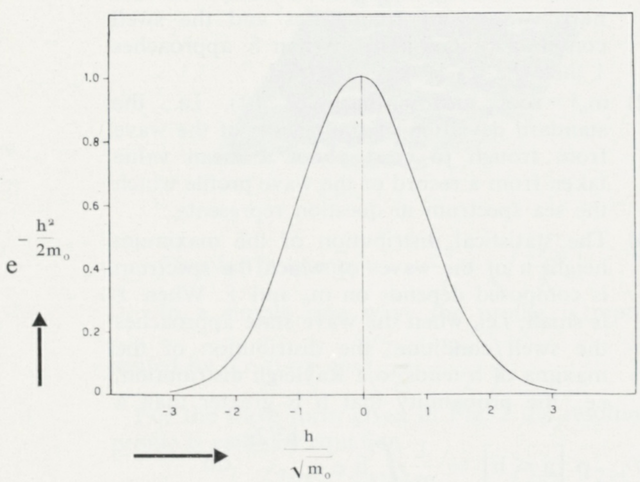

FIG. 5

Normal distribution

A wave spectrum based on theoretical considerations and experimental evidence which exhibits many observed wave properties, known as the Neumann spectrum, is often used. This spectrum coincides fairly well with certain spectra computed from actual observations, such as the

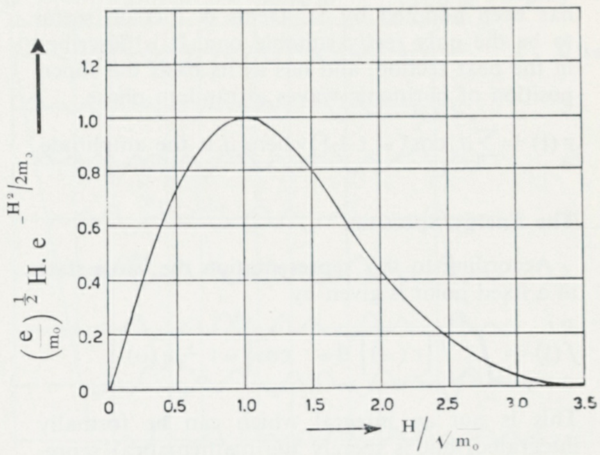

FIG. 6

Rayleigh distribution

Darbyshire spectrum, but there is still no unanimity among the authorities on the actual area of the spectrum or, indeed, on the shape, for defined wind conditions. In time it is hoped these differences will be reconciled and standard spectra will be available. In defence of the accuracy of the Neumann spectrum it can be said that it has been used with success to forecast the heights and periods of swells, but opinion in this country is tending towards accepting the Darbyshire spectrum as being more accurate. The energy spectrum denotes the state of the sea at a given point and for a period in which the spectra remain reasonably constant, say half an hour, and varies in shape and area with the wind conditions causing the storm.

There are three well-known wave spectra, Neumann, Roll/Fisher and Darbyshire, and they are shown in Figs. 7 and 8.

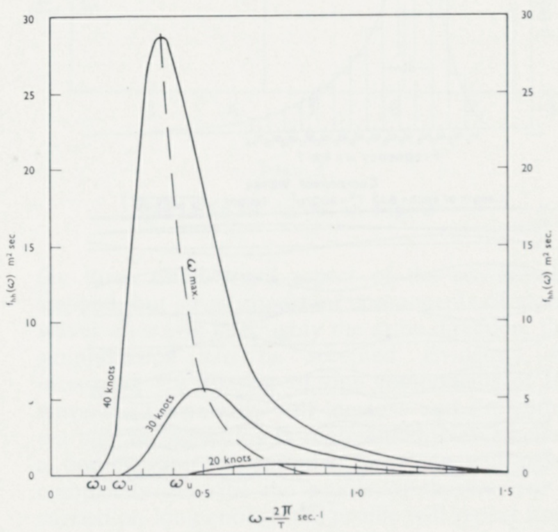

FIG. 7

Hypothetical spectral density as a function of wave frequency for ocean waves at different wind velocities

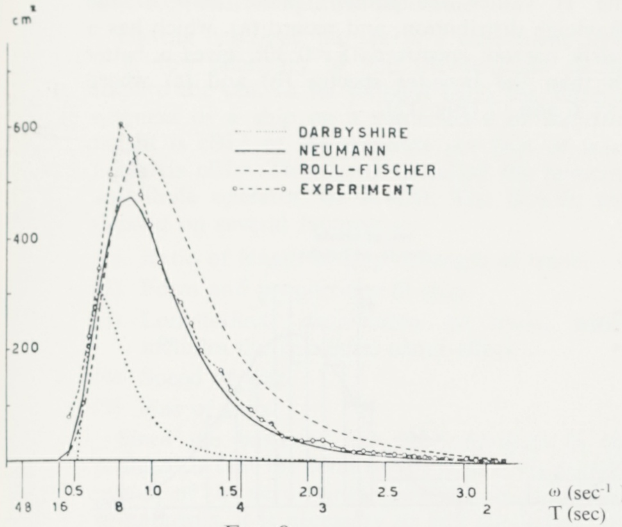

FIG. 8

Comparison of observed integrated spectral density with different hypothetical spectra for a wind speed of 18.7 knots

The ordinate of the Neumann spectrum has the following form:—

$$[r(\omega)]^2 = \frac{51\cdot1}{\omega^6}\, e^{\frac{-2g^2}{u^2\omega^2}} \text{ ft.}^2 \text{ sec.}$$

where u is the wind velocity in ft./sec. The value E, or $2m_0$, is $0\cdot242\left(\frac{U}{10}\right)^5$ ft.², where U is in knots.

Typical Neumann spectra are shown in Fig. 7.

These spectra refer to a fully developed sea, that is, a sea in which all components are present with their maximum amount of energy. For this the wind has to blow for a certain time, over a certain fetch, and there is still some doubt about the minima of these characteristics. The spectra obviously depend on the velocity of the wind, and fully developed storm seas at wind velocity of 50 knots and more are rarely encountered, because the required fetch and duration is far too long, except in localities such as round Cape Horn in the winter. The wind velocity governs the range of waves with an important amount of energy, and as the wind increases, the band containing the maximum energy is displaced towards the smaller frequencies, that is, the larger waves with the greatest periods assume an increasing importance. The frequency with the maximum energy is given by the formula $\omega_{max} = 15\cdot7/U$. Very long and very short wave components have little effect on the spectra, and do not dominate the wave profile, but the long waves do not decay so rapidly when the storm blows itself out, and they appear in the swell which follows the storm. Consequent on this, the swell spectrum is narrow and since the waves are reasonably regular, the swell lends itself to prediction of amplitudes much better than does the fully developed storm sea.

It may be useful to conclude this section with several typical wave records and their spectra to illustrate the foregoing. The wave records in Fig. 9 relate to:—

(a) Pressure records of a swell taken off Cornwall.

(b) and (c) Wave heights taken in the Bay of Biscay by a ship-borne wave recorder.

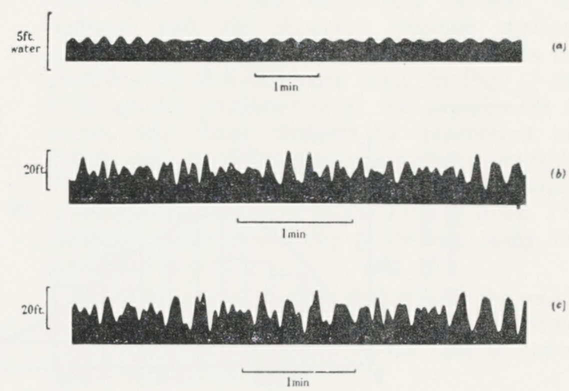

FIG. 9

Wave Record

An analysis of the components of these waves is shown in Fig. 10.

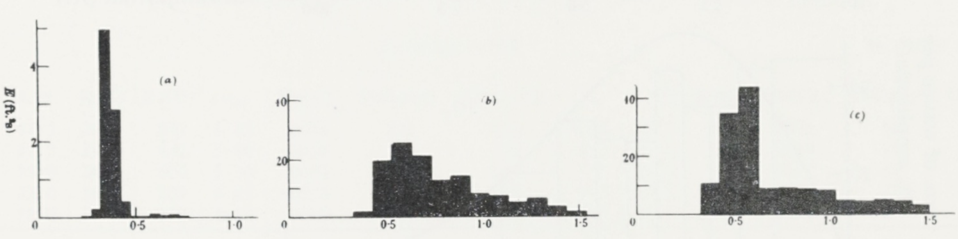

FIG. 10

Energy Spectrum

The narrow spectrum of the swell record should be noted.

The statistical distribution of h values is given in Fig. 11, where the approach to the normal distribution is evident. Finally, the distribution of the H values are shown; these tend to the Rayleigh distribution, and record (a), which has a fairly narrow spectrum ($\varepsilon = 0{\cdot}30$), gives a better fit than the broader spectra (b) and (c) where $\varepsilon = {\cdot}6$ and ${\cdot}7$ (Fig. 12).

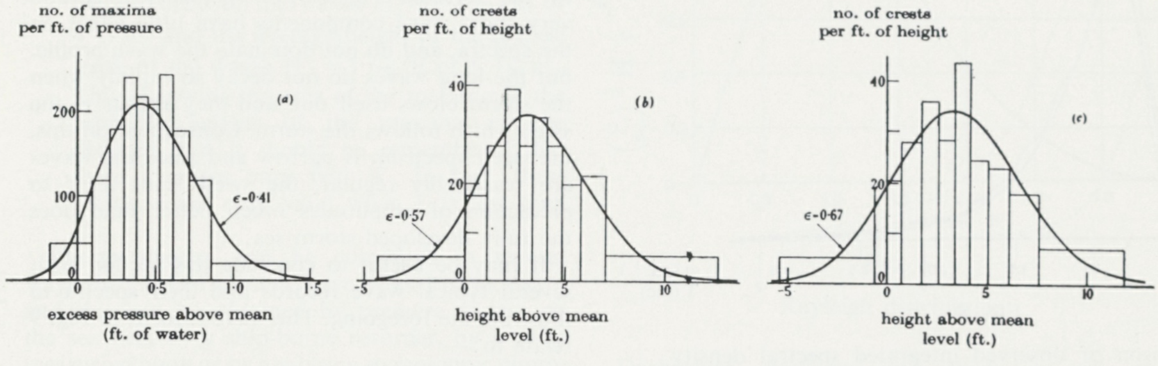

FIG. 11

Distribution of "h" values

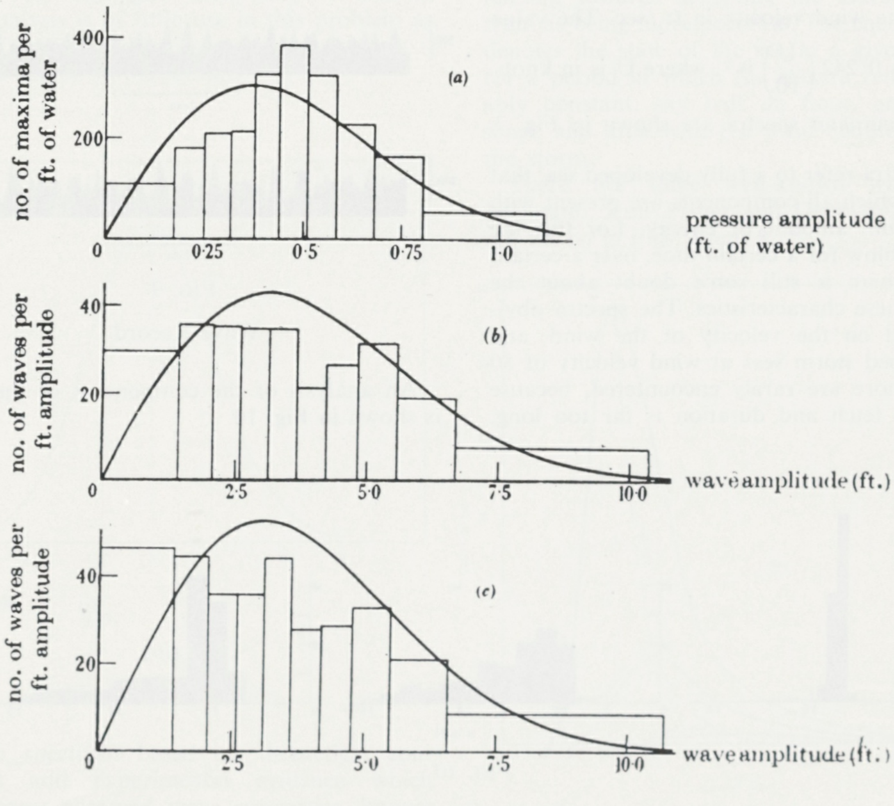

FIG. 12

Distribution of "H" values

The Response Amplitude Operator

Having described the sea in a convenient form it is now necessary to find the response of the ship to the sea. In its simplest form the bending moment of a ship on a static wave of standard height is cBL^3 or the response per foot of wave height is cBL^2. This factor is called the response amplitude operator, or R.A.O. The R.A.O. will depend on several factors:—

(1) Ratio of length of ship to length of wave.

(2) Form and proportions of ship.

(3) Longitudinal distribution of mass, which includes the entrained water effect.

(4) Speed of ship.

(5) Size of ship.

Since the basis of the new approach to the problem is that the total response to a sea is composed of a very large number of sinusoidal components, it is necessary to find the R.A.O. for each of these components and operate with it on the appropriate band in the energy spectrum to give the bending moment spectrum. This spectrum has the same properties as the sea spectrum, and therefore the probable B.M.'s can be found in the same way. The greatest B.M. the ship is likely to sustain can never be found by this method, the whole basis of which is that of probabilities. Incidentally, it may be observed that the probability basis is perhaps better adapted to determine pitching and heaving, for instance, than bending moments, since the naval architect is interested in the average range of pitching and heaving rather than extreme values.

The R.A.O. can be found in two ways, by calculation and by model experiments. The orthodox calculation can readily be made and the R.A.O. found for various ratios of wave length to ship length and block coefficients and account can be taken of the Smith Correction ; this calculation does not take into account the effect of speed and other important factors. Model results have shown fairly consistently, however, that further corrections have to be made which reduce the R.A.O. and more refined methods of calculation have been devised which take account of the effect of the ship on the wave as well as the effect of the wave on the ship, and also the longitudinal distribution of mass and the speed. It is becoming evident from studies of this kind that the wave bending moment is in fact related to the still water bending moment. Reasonable correlation between these calculations and model results have been obtained, but there is still much to learn, as the following example, included to illustrate the principles, demonstrates.

A destroyer model 5·71 ft. long was run at speeds of 0 and 2·53 ft. per second in regular waves 1·43 in. high and of lengths ·75L, 1·0L, 1·25L, 1·5L and 2·0L. The bending moments and hence the R.A.O.'s in these waves were found, and also calculated by the Jacobs method. Then the model was run in a confused sea at the same speeds and the sea and bending moment spectra (Fig. 13) found also. The bending moment spectra were then deduced by operating on the sea spectra by the R.A.O.'s found by experiment and by calculation. A close correlation between the deduced and the observed bending moment spectra was obtained if the calculated and not the experimental R.A.O.'s were used. In Fig. 13 the B.M. spectra predicted from the experiments in waves, and those obtained by experiment are shown, and the differences are quite appreciable. This circumstance indicates the uncertainty which is still attached to much of this kind of work. The following table shows the procedure, using the calculated R.A.O.'s.

It will be noticed that the R.A.O.'s for the speeds of 0 and 2·53 ft./sec. are practically the same. This would not necessarily be the case at higher speeds, nor for all ships.

V = 0								V = 2·53 ft./sec.					
						$Fn(BM)^2 \times 100$						$Fn(BM)^2 \times 100$	
$\frac{\lambda}{L}$	B.M.	RAO	RAO^2	ω_e	$Fn(\omega)$	Deduced	Observed	ω_e	RAO	RAO^2	$Fn(\omega)$	Deduced	Observed
·75	41·5	29·0	841	6·85	·0016	135	130	10·6	29·5	870	·0075	65	60
1·00	41·5	29·0	841	5·90	·0028	235	220	8·7	29·0	841	·0014	118	100
1·25	35·0	24·5	600	5·30	·0033	198	180	7·5	24·4	595	·0020	119	110
1·50	23·0	16·1	259	4·85	·0038	98	100	6·6	16·1	259	·0020	52	60
2·00	19·0	13·3	177	4·20	·0019	34	35	5·6	13·3	177	·0012	21	20

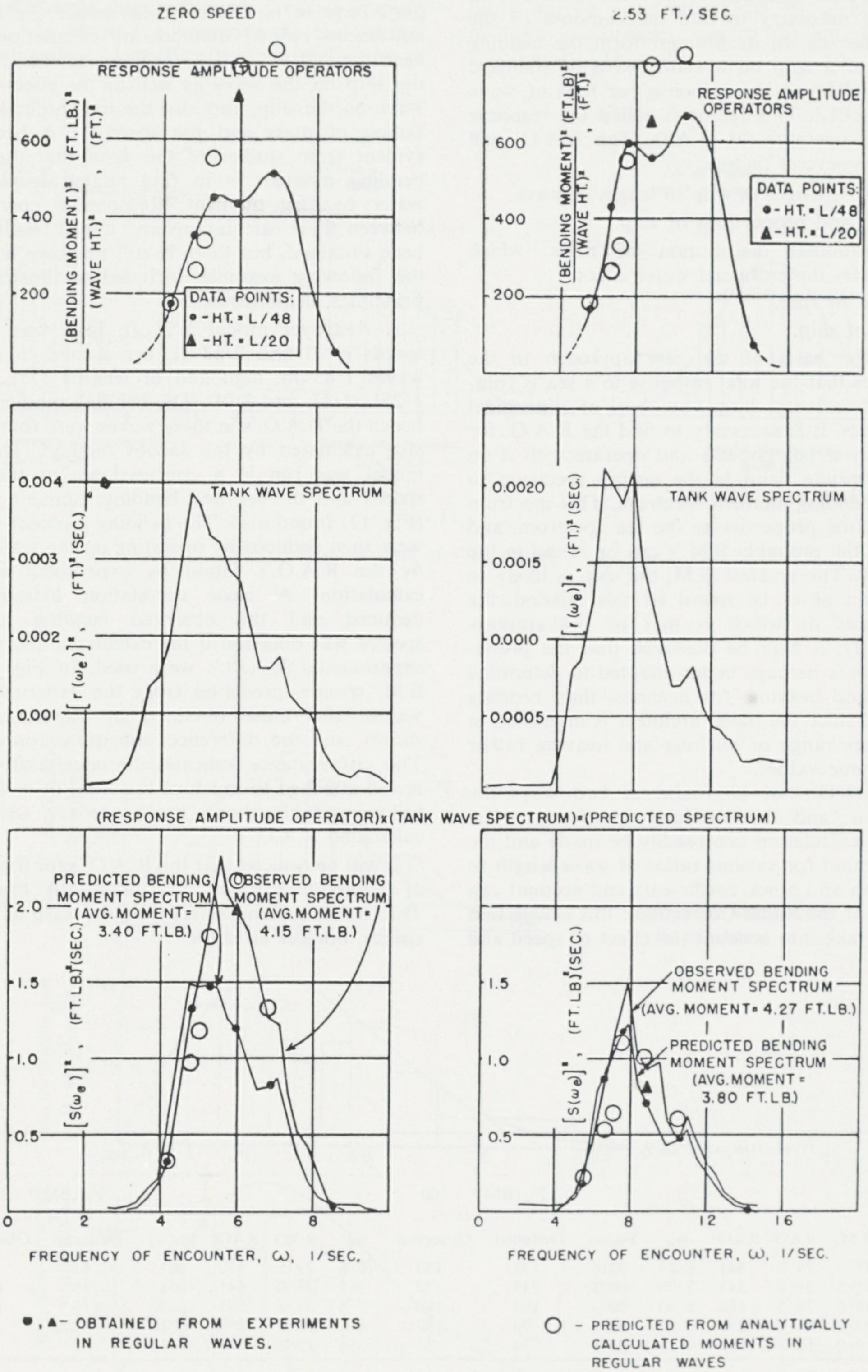

FIG. 13

Comparison of observed and predicted bending moment spectra for destroyer model 1723 in a high irregular sea

Effect of Speed on Energy Spectrum

The energy spectrum is referred to fixed co-ordinates in space, and therefore relates to the ship at rest. When the ship is moving the frequency of encounter changes, and the following correction has to be made. If T_e=period of encounter, ω_e = frequency of encounter and V=speed of ship:—

$$T_e = \frac{\lambda}{V+c}$$

$$\text{or } \omega_e := \omega\left(1 + \frac{\omega V}{g}\right)$$

$$\text{now } \frac{d\omega_e}{d\omega} = 1 + 2\omega\frac{V}{g}$$

$$\delta\omega_e = \left(1 + 2\omega\frac{V}{g}\right)d\omega$$

Conversely,

$$\omega = \frac{-1 \pm \sqrt{1 + 4\omega_e\frac{V}{g}}}{\frac{2V}{g}}$$

That is, as the speed increases, the length of each small element of the fixed spectrum increases, but the area is constant and as Fn (ω) dω =Fn (ω_e) dω_e, the total area of the spectrum remains unaffected by this operation, which is to be expected as the energy content of the sea does not depend on the speed of the ship. In Fig. 13 is shown tank energy spectra for a speed of 0 and 2·53 ft. per sec. There, the area of the two spectra is equal.

Effect of Size

It is necessary to consider the effect of size on R.A.O. so that the relation between two ships of different sizes but of the same geometric characteristics can be obtained. R.A.O. is proportional to L^3 when $\frac{\lambda}{L}$ and in a Froude system $\frac{V + c}{\sqrt{L}}$ are constant. The first condition is self-evident and the second follows from the following:—

Let $L^1 = n\,L$

$\lambda^1 = n\,\lambda$

Then $c^1 = c\sqrt{n}$ and

$$\frac{V^1+c^1}{\sqrt{L^1}} = \frac{(V^1 + c\sqrt{n})}{\sqrt{n\,L}} = \frac{V+c}{\sqrt{L}}$$

substituting V^1 and λ^1 into $\omega_e = \sqrt{\frac{2\pi g}{\lambda}} + \frac{2\pi V}{\lambda}$

leads to $\omega^1{}_e = \frac{\omega_e}{\sqrt{n}}$

From this relation, if R.A.O. for a ship L and speed V is known, the R.A.O. for a ship nL and speed $V\sqrt{n}$ can be obtained by transferring the R.A.O. appropriate to ω_e to $\frac{\omega_e}{\sqrt{n}}$. Thus, as the ship becomes larger the R.A.O. band is contracted for the same sea spectrum. Furthermore, $\omega^1 = \frac{\omega}{\sqrt{n}}$ so the frequencies in a sea spectrum are much smaller than those in a tank spectrum and therefore the relation between ship and model is maintained.

Practical Application

Having found the B.M. spectrum and hence the probable average B.M.s how much further on are we than before? Certainly, from these new methods we should know the effect of speed and have a better understanding of the other factors involved. Furthermore, if the sea spectra for different oceans under different conditions are known, the corresponding B.M.s likely to be sustained under these conditions will also be known. Superficially, then, we are back where we started, for so far as the B.M. amidships is concerned, any sea spectrum can be replaced by an equivalent trochoidal wave, and indeed it has been shown that making certain rather wide assumptions the effect of a sea represented by the Neumann spectrum for a wind of 50 knots on a ship hove to is equivalent to that of a wave having a height of $1{\cdot}1\sqrt{L}$ for lengths of about 400 to 900 ft. This is perhaps a fortuitous justification of the L.R. practice, but it is none the less welcome for that. However, there is more to the matter than that. To arrive at a proper standard we would need a very large number of equivalent waves which would vary with the length of ship, the speed, the form, the distribution of weights in the ship, the weather conditions and the oceans traversed, and other factors.

Another Approach to the Subject

A further extension of the use of probabilities in determining the forces on a ship has been made by Jasper. He has shown that if a short-term distribution of wave heights and also stresses—that is, the distribution over a period of, say, half an hour when conditions are steady—is Rayleighan, then the long-term distribution—that is, the distribution over several months or years when the same kind of conditions apply—is log normal. That is, if the wave heights or stresses are plotted on log normal probability paper they will fall on a straight line. This data can be obtained from records taken of the height of waves by weather ships, and from records of stresses from self-recording strain gauges fitted to ships in service. Jasper's distribution of wave heights for the North Atlantic is given in Fig. 14 and it may be interpreted in the following way.

Let it be assumed that the 99·0 per cent probability point is taken, that is, the significant height of wave is 30 ft.; this means that over a very long period one wave out of 100 is over 30 ft. high. From this the correspondng wave spectrum can be determined, always assuming that the Neumann spectrum is valid.

$30 = 2{\cdot}83\sqrt{E}$ or $4\sqrt{m_o}$

$\therefore\ E = 112 = {\cdot}242 \times \left(\frac{U}{10}\right)^5$ where U is wind speed in knots. From the Neumann spectrum, this E value is that for a sea generated by a wind of 34 knots blowing for 45 hours.

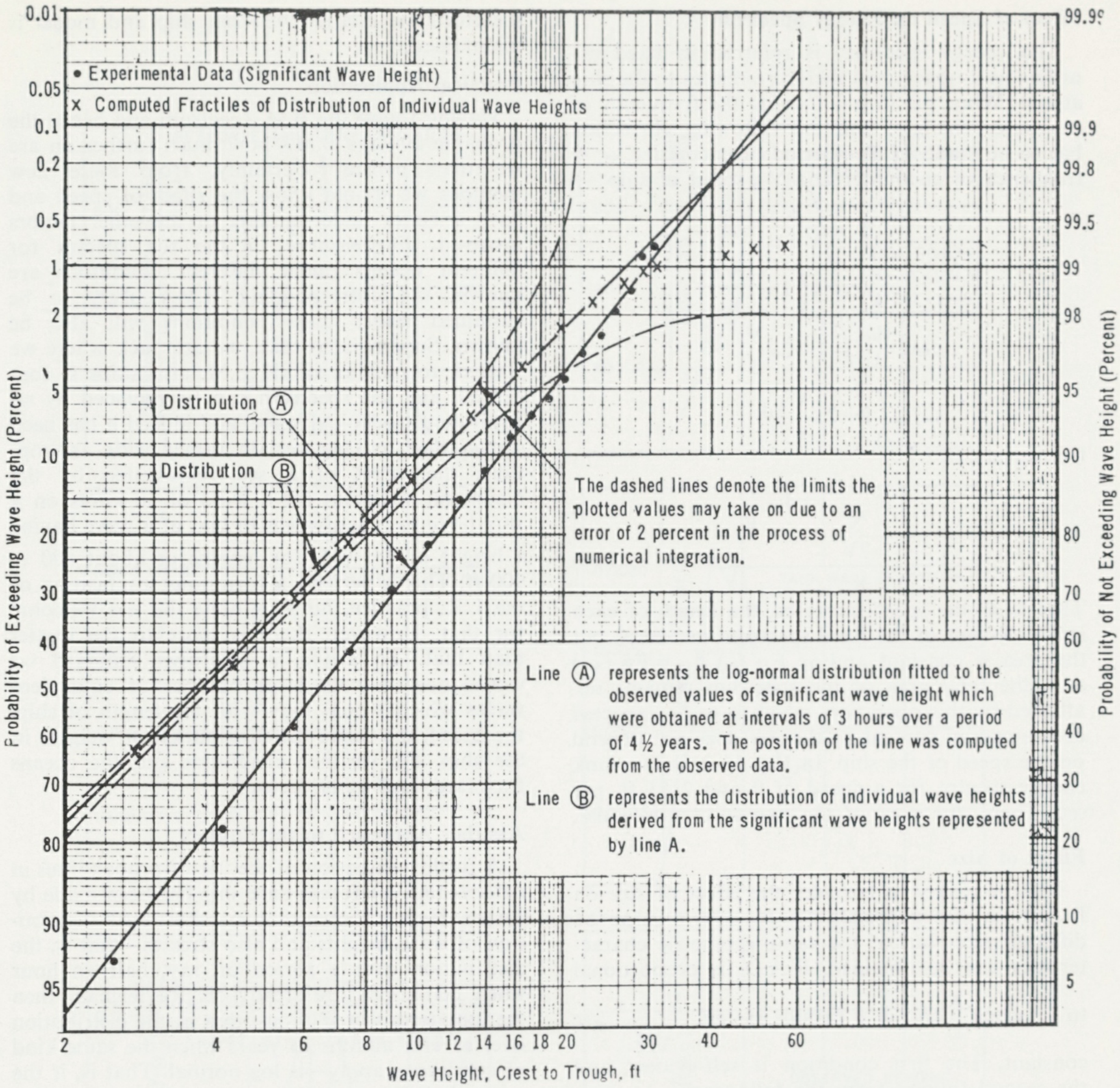

FIG. 14

Distribution of the Heights of Ocean Waves at Weather Station C, 52° N 37° W, North Atlantic Ocean

(This distribution is based on 12,365 observations made over a period of 4½ years by U.S. Weather Bureau personnel)

Then if the R.A.O.'s were known the corresponding B.M. spectrum could be found. From that, of course, the equivalent wave could also be found. If data were available this procedure could be applied to other oceans, and in time a series of appropriate equivalent waves relating to a standard assumption could be determined.

Alternatively, the hull girder stresses could be taken as a basis, and in Fig. 15 Jasper's analysis of various ships is shown. For example, in the *Ocean Vulcan* the observed stress at 99 per cent probability has a range of 1·5 tons/in.[2]. Similarly data would be required for other sizes and types of ships at different speeds and in different services.

The *Ocean Vulcan* results are of particular value as they relate to a ship sailing on the same route at a reasonably constant speed for several years.

Jasper's work has been extended by Kalbfleisch and Bonizec who have indicated a method of determining the maximum probable stresses from the log normal distributions.

It would be unwise to think that this whole subject is now cut and dried; in fact, there are many discrepancies which have to be resolved. The log normal distributions shown in Fig. 15 are not very convincing at the higher stresses, and it is difficult to reconcile the various lines. Stress distributions for two other ships have been made available to L.R. and have been added to the diagram. Ship A traded on the North Atlantic and

therefore should have fallen more closely in line with the *Ocean Vulcan* especially as the theoretical static R.A.O.'s were similar. The speed was higher and this may be an important factor. Ship B is more in accordance with the *Ocean Vulcan* results but was engaged in a much less turbulent area. Another matter requiring more elucidation is the distribution of the higher stresses. In Fig. 16 the log normal curves have been plotted on probability paper extending to ·0001 per cent probability or one chance in 1,000,000. It will be observed that the spots for Ship A conform very well with the log normal line at the very low probabilities, but those for Ship B do not.

Finally, it must be emphasised that these approaches to longitudinal strength are still in their infancy, but there is no doubt that in the future they will replace the methods used at present. Nevertheless, the isolated large wave of regular form which will appear sooner or later must always be kept in mind.

The Author would like to acknowledge the help given by Mr. McCallum and Mr. de Wilde in the preparation of this paper.

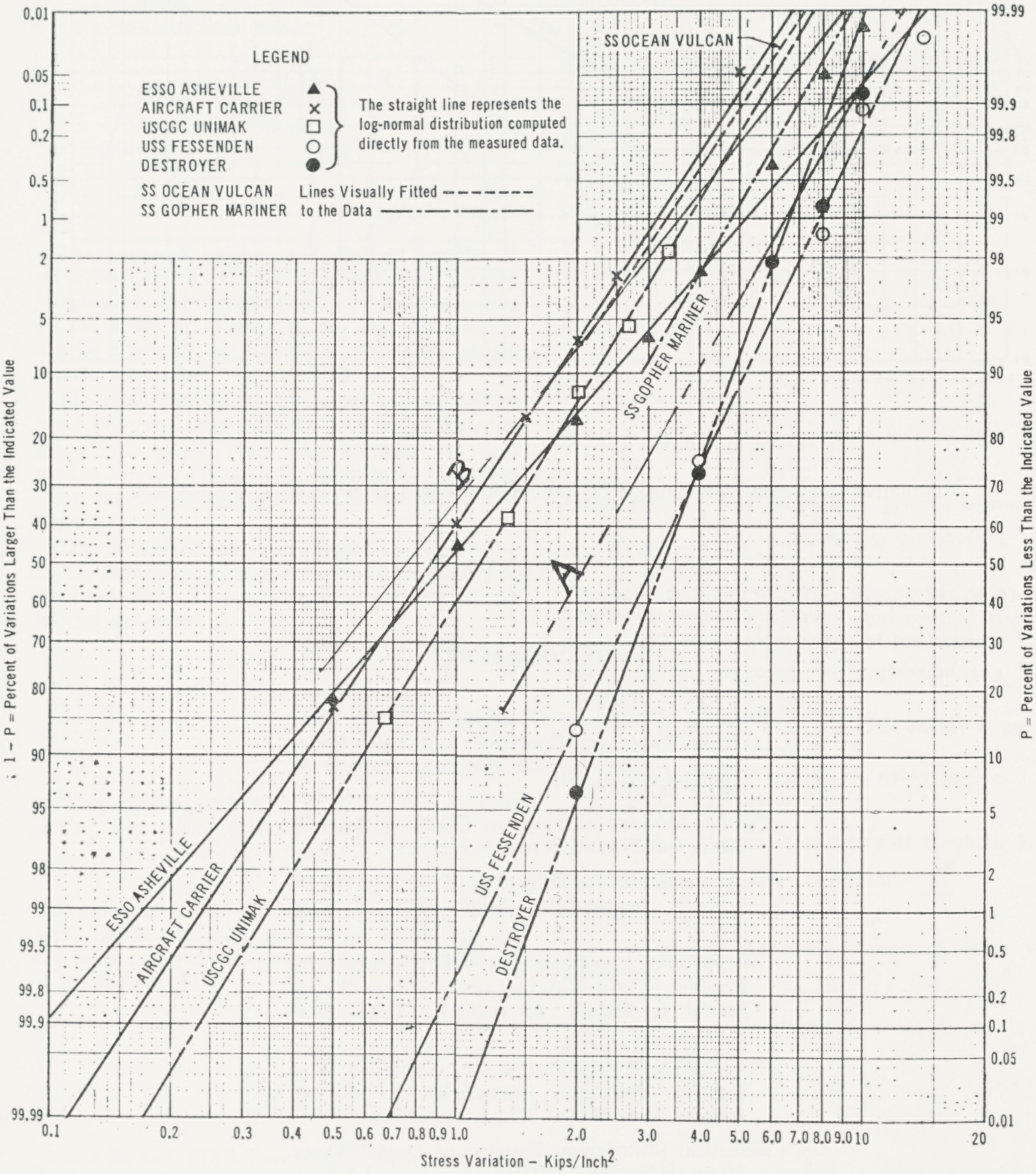

FIG. 15

Cumulative "Long-Term" Distribution of Wave-Induced, Hull-Girder Stresses for Several Ships

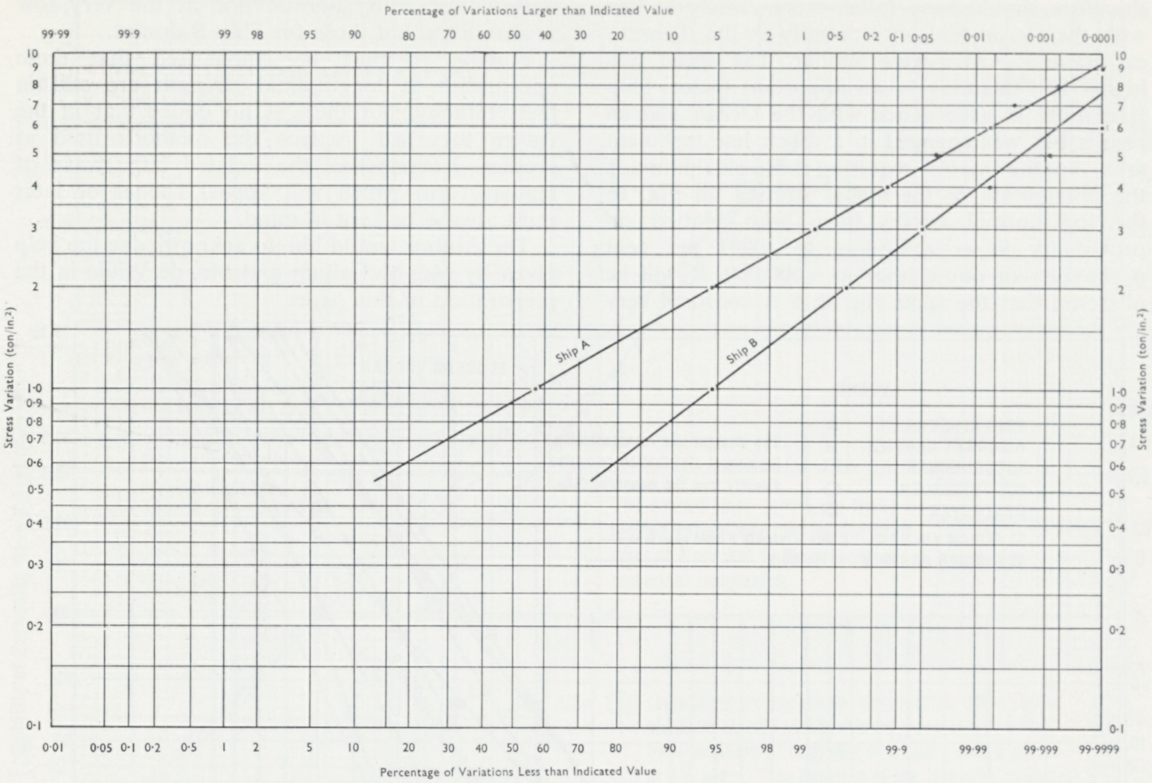

FIG. 16

Cumulative "long-term" distribution of wave-induced, hull-girder stresses, ships A & B

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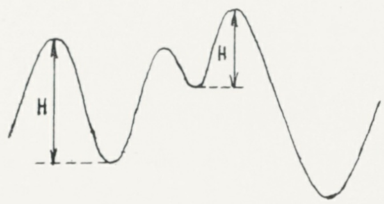
APPENDIX

DERIVATION OF NEUMANN SPECTRUM

R = energy in spectrum
T = period
ω = frequency
ρ = density of water
H = wave height
λ = wave length
g = gravitational acceleration

} apparent

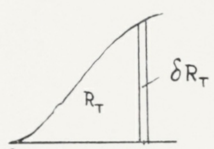


The apparent heights, periods and lengths of waves in a seaway can be measured, and they can be related to the energy of the seaway, as described by the spectrum. Furthermore, their distribution can be predicted, on a basis of probability as has been mentioned.

The derivation of the Neumann spectrum from these observed characteristics and from theory is given in a paper "Zur Characteristik des Seeganges", and it is fair to say that most published work on the energy spectrum is based on this paper.

The outline of the basic derivation follows:— In the energy spectrum, where R_T is the energy in the limits O to T and F(T) is the ordinate of the energy spectrum.

$$F(T)=\frac{\delta R_T}{\delta T}$$

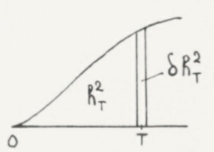


and the spectral energy in the band $T-\frac{1}{2}\,\delta T$ to $T+\frac{1}{2}\,\delta T$ is defined approximately by

$$\frac{\delta R_T}{\delta T}=\frac{1}{8}\rho g\frac{\delta h_T^2}{\delta T}$$

The wave can also be defined by means of the wave height spectrum, and if $h_T^{\,2}$ is the sum of the squares of the wave heights in the spectrum in the limits O to T,

$$(H_T)^2=\frac{\delta h_T^2}{\delta T}$$



Now $\delta R_T = F(T)\,\delta T = \frac{1}{8}\rho g\frac{\delta h_T^2}{\delta T}dT$

and the total wave energy in a storm sea

$$R=\frac{1}{8}\rho g\int_0^\infty\frac{\delta h_T^2}{\delta T}\,dT=\frac{1}{8}\rho g\int_0^\infty H_T^2\,dT$$

On the assumption that

$$\frac{(H)}{(\lambda)_T}\sim e^{-(gT/2\pi U)^2}$$

where U is the velocity of the wind in consistent units.

and since $\lambda=\dfrac{gT^2}{2\pi}$

Then $H_T^{\,2}=C\dfrac{g^2T^4}{4\pi^2}\,e^{-2\left(\frac{gT}{2\pi U}\right)^2}$

and $\delta R_T=F_T\,\delta T=+C\rho\dfrac{g^3}{32\pi^2}T^4\,e^{-2\left(\frac{gT}{2\pi U}\right)^2}dT$

Transferring to frequency ω

$F_\omega\, d\omega = F_T\, d\,t$ and $\delta R_\omega = F_\omega\, d\omega$

$$d\omega=-\frac{2\pi}{T^2}\,dT$$

$$\delta R_\omega=F_\omega\,d\omega=-\,C\rho g^3\pi^3\,\omega^{-6}\,e^{\frac{-2g^2}{\omega^2U^2}}\,d\omega$$

$$R_\infty^\omega=\int_\infty^\omega\delta R_\omega=-C\rho g^3\pi^3\int_\infty^\omega\omega^{-6}\,e^{\frac{-2g^2}{\omega^2U^2}}\,d\omega$$

Integration of this expression gives the result

$$R_\omega^\infty=C\rho g^3\pi^3\left[\begin{array}{l}F_n\text{of a probability distribution}-\\F_n\text{of wind speed}\end{array}\right]$$

This solution links with the distribution of wave heights implicit in the theory, and permits of the use of expressions for maxima of the heights.

For practical purposes, the expression can be expressed as follows:—

$R=C\rho\,3{\cdot}65\,g^{-2}U^5$, i.e., the total energy is proportional to (wind speed)^5

From observation $C=8{\cdot}27\times10^{-4}\ \text{sec}^{-1}$

$\therefore R=3{\cdot}125\times10^{-9}U^5$

It is usual to express R in terms of E where E the area of the amplitude spectrum $=\dfrac{2R}{\rho g}$

$\therefore E=6{\cdot}4\times10^{-12}U^5\ \text{cm}^2$

The ordinate of the energy spectrum curve is therefore

$$F(\omega)=\frac{2C\rho g^3\pi^3}{\rho g\,\omega^6}\,e^{\frac{-2g^2}{\omega^2U^2}}$$

$$=4{\cdot}8\,\omega^{-6}\,e^{\frac{-2g^2}{\omega^2U^2}}\ \text{cm.}^2\times\text{sec.}$$

$$\text{or }51{\cdot}1\,\omega^{-5}\,e^{\frac{-2g^2}{\omega^2U^2}}\ \text{ft.}^2\times\text{sec.}$$

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Discussion

on

Mr. J. M. Murray's Paper

THE NEW APPROACH TO LONGITUDINAL STRENGTH

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

Discussion on
Mr. J. M. Murray's Paper

The New Approach to Longitudinal Strength

Mr. J. E. BEKEN

There is one small point I would like Mr. Murray to clarify.

On page 7 he states "It is becoming evident from studies of this kind that the wave bending moment is in fact related to the still water bending moment".

In previous papers by Mr. Murray I believe "wave bending moment" has referred to the superimposed bending moment due to the passage of a wave. In this paper "wave bending moment" appears to mean total bending moment in the trough of a wave. Is this correct?

Further, it has long been established that there is a relationship between the wave bending moment and the still water bending moment governed by the form of the ship, but I presume that the reference in the paper concerns a change of still water bending moment in a given ship, arising from a different distribution of cargo.

If this is so, does it follow that where N_2 is less than N_1 some reduction may be made to the minimum value for N?

(This refers to the numeral for obtaining required deck and bottom area in tankers.)

Mr. G. M. BOYD

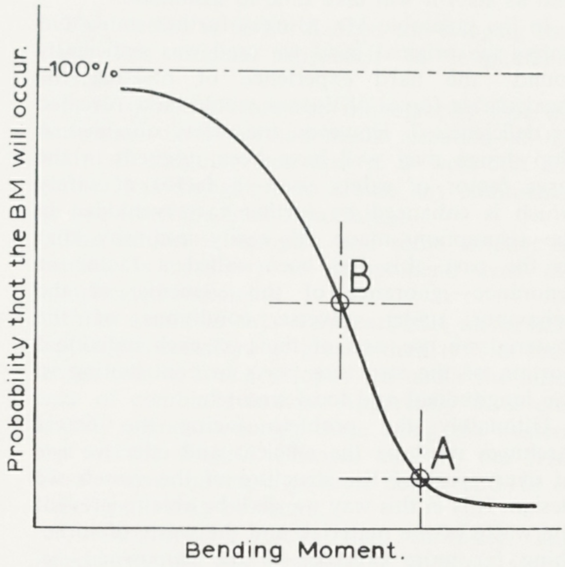
The paper reflects the strong tendency towards complication that is occurring in many branches of technology. This is of course necessary in most cases, and particularly in this one, since the phenomena we are studying are very complex. The simple methods of a generation ago are no longer adequate.

It also reflects the growing trend towards the statistical approach to design. It is no longer respectable to say that a structure is perfectly safe up to a certain load, and unsafe above it. One can only say that up to a certain load, the risk of failure is acceptable.

On page 9, the Author asks: "How much further on are we than before?".

The question is of course unanswerable, but it appears that these methods provide us with a more rational picture of what is actually happening. It is easier to see where we are heading, and that seems to be towards a more accurate estimation of the bending moments to which a ship will be subjected during her lifetime.

This might be expressed by a diagram such as Fig. 15, which translated into more familiar form would appear as follows:—



In this diagram, we see that the frequency, or probability of high bending moments diminishes, but does not become zero.

The usual approach is to select an acceptable probability, say 2 per cent or 5 per cent, note the bending moment that has this probability (point A on diagram) and design the structure for this bending moment.

It may be, however, that some smaller bending moment which occurs more frequently (point B of diagram) may be more damaging, and more significant. This is a question which is engaging many structural designers concerned with fluctuating loading, and a good deal of further data on the point seems desirable.

Having found the significant bending moment, the problem arises of how to design the structure to withstand it. This problem is in the domain of strength of materials, in which field the ignorance is almost as abysmal as that in the field covered by the paper. This other field is also in a state of revolutionary change, and it is becoming increasingly evident that the simple concepts of stress which have served so well in the past are no longer adequate.

Referring to Fig. 2, it seems that once a typical wave profile is established in this way, the next step would be to establish the changes of wave profile which are likely to occur within the actual length of the ship. Anything that occurs outside this length is irrelevant. Presumably this is taken into account in computing the Response Amplitude Operator as indicated on page 7, but it is by no means clear how this and the other factors listed are allowed for.

Mr. L. J. BRINTON

On first reading this paper one may feel rather like a bee trying to fly round in a barrel of treacle—expending much effort and not getting very far; but as Mr. Murray states this is a new approach to the problem of longitudinal strength and as such it will take time to assimilate.

In the preamble Mr. Murray further states that unless the original basis we used was sufficiently sound "the hard experience of meeting the incalculable forces of the sea would have revealed its deficiencies". However, the safety obtained in ship design may well have been inherent in the large factor of safety used—a factor of safety which is enhanced by further factors hidden in the assumptions made. It's easily seen why, that in the past, this has been called a factor of ignorance—ignorance of the elements, of the behaviour under adverse conditions of the material we use, and of the part each individual portion of the structure plays in contributing to the longitudinal and local strength.

Ultimately the problem facing the Naval Architect must be the efficient and effective use of every part of the structure of the vessels we design. It is in this way we shall be able to prevent the waste of the material, and the waste of something far more precious to the shipowner, i.e. deadweight. It is in this direction that Lloyd's Register can take its proper place in leading the field of structural design and this paper, revising as it does our approach to longitudinal strength, must be a welcome step forward.

Obviously the proof of this new approach must be the facility with which the results can be employed and perhaps Mr. Murray could tell us whether these are now being regularly used in L.R. If so, it would be interesting to know whether the actual work involved is simpler than would appear from the paper, in which case a child's guide of practical application would be very useful for those of us who have not made any approach whatsoever to longitudinal strength for some time.

After having had my brain turned upside down by reading this paper I can sympathise with the printer who appears to have had the same trouble in typesetting ω in the first column on page 3, in the function $\sqrt{[r\ (\omega)]^2 d\omega}$.

Finally, I would like to thank Mr. Murray for his presentation of this paper this evening, because his explanatory remarks answered some at least of the questions in my mind.

Mr. J. McCALLUM

Mr. Murray has an inordinate facility for whittling the chaff from a problem and reducing the most complex concept to manageable proportions. This review of the comparatively new science of ship responses in ocean waves is so compact and logically presented as almost to beguile the unwary into believing that the problem can be completely defined in a few pages. The Author himself, however, leaves us in no doubt that there are many discrepancies remaining to be resolved.

The various spectra which have been developed to represent sea states have been the subject of spirited discussion on both sides of the Atlantic and in Japan. In fact, there may be more agreement than is readily apparent, for the suggestion has been put forward that the Neumann spectrum is a fairly accurate representation of sea states off the American Atlantic seaboard, while the more strongly peaked Darbyshire spectrum represents the type of sea experienced in European waters. There might well be a difference in the amplitude-frequency relationship which is attributable to local features such as the Atlantic shelf or Gulf Stream currents. Both spectra are empirical represensations—the Neumann much neater mathematically than the Darbyshire, but American confidence in the Neumann spectrum has been considerably shaken by recent observations round the northern shores of Britain.

In calculating the wave spectrum, manual manipulation of Fourier series is of little use (as Mr. Murray points out) as the low order harmonics may be neglected. This is, however, the method used by digital computer processes. Successive values of the wave record ordinate at frequent intervals are fed to the computer on data tape, and a Fourier analysis of a high order is performed, either directly or in stages.

As in the development of most theoretical work, a number of different sciences with varying terminologies have become involved. Naval architects, ship model tank experimenters, statisticians, oceanographers and mathematicians have all had a part to play, and some confusion of definition of the main parameters still exists. In particular, it would be of value to reach common agreement on symbols. Height and half-height of wave forms are frequently represented by the same symbol, h (or H). The consistent use of a for amplitude as in this paper would eliminate one source of misinterpretation. Again, the representation of the wave state adopted by St. Denis and Pierson, $\int_0 \cos\,[\omega t+e(\omega)]\ \sqrt{[r(\omega)]^2 d\omega}$ is really a multiple ($\sqrt{2}$) of the wave state corresponding to the amplitude component of the energy in a wave, $\frac{1}{2}a^2$. We are thus saddled with an obtrusive 2 in the energy spectrum ordinate in all work based on St. Denis and Pierson's original paper. No doubt a common language will evolve as time goes on.

This statistical approach will undoubtedly have a strong influence on future longitudinal strength calculations, and Mr. Murray is to be congratulated on the timely presentation of a thoughtfully constructed and comprehensive synopsis of a relatively new concept.

Mr. J. B. DAVIES

It is frequently difficult to appreciate, when a paper puts forward a new concept, whether or not it will eventually become a matter of importance. By the time it is evident that it will have an effect

on one's normal work, the practitioners in the speciality have built up a code of terminology and agreed assumptions which often makes it difficult to understand what they are doing.

During the eight years which have elapsed since the original paper was written by St. Denis and Pierson, the practitioners in this somewhat abstruse branch of the art or science of naval architecture have developed their own terminology, and we must be very grateful to Mr. Murray for giving us this summary of the present-day position.

I was very pleased to see the Author quoting Professor Lewis's remark that some of the simpler concepts can be applied without a detailed understanding of the methods, since much of the mathematics involved is of a considerably more advanced level than that normally covered in a naval architect's training. Again, some knowledge of statistical methods would appear to be necessary, and this also is a subject foreign to most of us.

However, whether we can understand the basic mathematics or not, there can be little doubt that we will have to live and work with the methods and concepts described in this paper, and it therefore behoves us to get up to date as quickly as possible.

Much of this progress has been due to collaboration between the naval architect and the oceanographer, and the development of suitable instruments for measuring both the actual wave patterns and the resulting ship (or model) stresses. This work is continuing in various countries and a most welcome degree of international co-operation has been achieved.

Most of us will probably be mainly interested in the question of the practical application of these methods. It would be foolish to pretend that they will lead to an *immediate* reduction in scantlings or redistribution of material, but they should lead to a much better understanding of the problems involved. This may well be of particular value when dealing with designs departing from the normal or for ships intended for service in a particular sea area. It is, I think, generally realised that the assumptions we now make are only approximations limited by our present knowledge, and while we may say that it does not matter how accurate they may be since they are only used as a basis for comparison, there can be no doubt that the more accurate we can make our assumptions, the more realistic will be our comparisons.

Mr. R. M. HOBSON

It would appear that specialised information and records will have to be taken into account when determining the scantlings of a ship for service in a particular area.

Such information may not be available to the design staff of small shipyards, or if available it may not be interpreted exactly as the Society may wish.

In view of these factors, it seems probable that if sea spectra are to be taken into consideration, then the Ship Plans Department will find itself called upon to stipulate the required "minimum scantlings" for many more cases than at present. This will result in a considerable increase of work for the Department.

Mr. R. G. LOCKHART

The Author is to be thanked for bringing forward this original approach to longitudinal strength.

It is apparent from the Bibliography that the Americans have devoted much attention to the study of irregular seas and it is well that the Author has thought fit to arouse any enthusiasm he can from the Staff Association.

The subject is complex and some amount of reading is necessary to comprehend even the complicated rudiments.

From available data it appears that waves having heights considerably greater than 1/20 of their length can be found in open sea under the influence of combined storms. Apparently in several known regions, amongst these being the western end of the English Channel and off the Irish Coast, waves having a possible height of 1/10 of their length may be found.

Oceanographers report that confusion and irregularity are the features of a wind storm. Fluctuations of wave height is a signicant feature of the irregular wave.

An energy spectrum analysis as described by the Author provides the basis for establishing the character of the pattern of wave heights by specifying the relative amplitude of the component waves.

Amongst tests carried out by the Americans in conditions equivalent to irregular seas are a series on a model of a T 2 Tanker. This particular type being chosen because of the known structural defects and the available data.

The 503 ft. ship was represented by a model 4·79 ft. long. Tests being carried out in regular and irregular waves.

It appears that the maximum bending moment occurred in irregular waves when the model encountered a wave of about its own length and 4·9 in. high this giving a ratio of about 1/12 and producing a moment more than double that corresponding to a regular wave of the same length. The general conclusion was that the increase in bending moment was partly due to the wave height and partly to the difference in dynamic effect. However intermittently such waves may occur, the magnitude of the bending moment is interesting and the Author is requested to comment on this with particular reference to actual ship behaviour.

The writer can find no evidence of any such tests being carried out to correspond to a ship in ballast and at varying drafts. The thoughts of the Author would be welcomed on this.

As could be supposed slamming appeared to be more severe in irregular waves and a large and suddenly applied sagging moment occurred at the instant of slam.

The writer is of the opinion that a reduction in speed is called for when this occurs and also considers that any tendency to brittle fracture is probably at its height on such occasions.

It is to be hoped in these days of computer programming that this country will not lag behind, because undoubtedly the method briefly described by the Author is the only realistic approach to the analysis of the behaviour of a ship at sea. While a great deal of data is required the sooner a start is made to collecting it the more correct will become the approach to longitudinal strength.

Mr. G. DE WILDE

It has always been Mr. Murray's object, through his Staff Association papers and reprints from his papers to technical institutions, to keep the staff informed about what is being done in longitudinal strength. To-night Mr. Murray tells us about some of the new ideas that are at the moment being applied to longitudinal strength.

The two most important developments seem to be that firstly, the profession has slowly started to realise that loads of the kind that a ship experiences, alternating loads of a random nature, can only be described with the aid of probability theory and, secondly, that naval architects and oceanographers had a common interest, the sea, and that co-operation was essential if we were to get nearer to solving the problem of the strength of ships. Fortunately, other branches of science, i.e. communications, were also dealing with random phenomena and it is from the concepts which originated there that St. Denis and Pierson developed their theory.

There are one or two comments I would like to make. On page 4 under (4) various crest to trough wave height figures are given as $C\sqrt{E}$. The constants were obtained by doubling the theoretical values of the constants for wave amplitudes. For the high wave heights this means that it is assumed that an exceptionally high crest is always followed by an exceptionally deep trough. This is clearly not always the case, the probability of a certain event occurring twice in succession must be lower than the probability of that event occurring on its own. Although there is no theoretical justification for it, it is perhaps true to say that a high crest is more likely to be followed by an average trough than by a deep trough. Using this, the average tenth highest wave is $2{\cdot}69\sqrt{E}$ instead of $3{\cdot}6\sqrt{E}$. Wave height records could provide experimental evidence one way or the other.

Under "Practical Applications" Mr. Murray points out that the effect of a sea represented by a 50 knot Neumann spectrum is equivalent to that of a $1{\cdot}1\sqrt{L}$ wave. This needs some further qualification because the effect of a sea can only be expressed in statistical quantities such as the mean of the third highest or tenth highest wave. At just under 600 ft. the $1{\cdot}1\sqrt{L}$ wave coincides with the average of the third highest waves and at just over 500 ft. with the average of the tenth highest waves. These results which were obtained using static R.A.O.'s were published by Akita and were confirmed by calculations done in the Society.

Incidentally, when the reduced coefficient $c=2{\cdot}69$ is used for the tenth highest wave instead of $c=3{\cdot}6$, then the $1{\cdot}1\sqrt{L}$ wave coincides with the tenth highest wave at just over 600 ft. Much more important, however, than which statistical prediction from a 50 knot spectrum the $1{\cdot}1\sqrt{L}$ agrees with at a particular length, is the fact that between 300 ft. and 700 ft. $\sqrt{L}$ is not directly proportional to $\sqrt{E}$. Above 700 ft., however, $\sqrt{L}$ is fairly proportional to $\sqrt{E}$. The $\sqrt{L}$ curve can, of course, never be divorced from the curve of acceptable stresses. Our acceptable stresses increase with length of ship. There is a case for having acceptable stresses varying with length because of the corrosion allowance which is a constant amount rather than a constant percentage, and also because complicated tri-axial stress conditions are more likely to occur in the heavier plating of the larger ships. It is also possible that there is a scale effect in the permissible stresses for two geometrically similar structures. It is unlikely, however, that the whole increase in acceptable stress between 300 ft. and 900 ft. can be explained in this way, so at the moment the acceptable stress is a convention which must be considered in conjunction with the wave height. If the wave height were adjusted to give a more constant stress, then this would mean reducing the slope of the wave height curve and to make it proportional to $\sqrt{E}$ would require an increased slope.

Fig. 14 in Mr. Murray's paper gives basically the probability distribution of E values, so giving the frequency of occurrence of sea states, when this is the case, then assuming the validity of the Neumann spectrum the frequency of occurrence of any spectrum is known.

It must be realised, however, that from a plot of individual wave heights such as obtained from statistical wave height metres, no conclusions about the frequency of occurrence of spectra can be drawn. Fig. 14 was the result of visual observation of wave heights, the heights of individual waves could of course not be assessed and a "significant wave height" was reported which is, in fact, a characterisation of the sea state, and so is a measure for $\sqrt{E}$.

Fig. 15 is rather reassuring in that the lines tend to converge at the low probabilities. In its present form it is difficult to draw any general conclusions from it because the sizes, section moduli, speeds, forms, mass distributions, etc., differ for the various ships. To eliminate at least some of the variables to some extent from the comparison, the lines could be converted into equivalent wave heights by multiplying the stresses by the section modulus and dividing by the static R.A.O.'s $h=\frac{\sigma\times W}{CBL^2}$.

Mr. O. M. CLEMMETSON

The Author has set out for us in this paper the foundations of a vast amount of work which is at present being done on waves. As in many other fields of science, oceanography is advancing very rapidly and, although one may take a small amount of comfort in the Author's statement that much of the mathematics of the subject is beyond the capability of most naval architects, it behoves us to have at least a general understanding of what is being done and to be able to interpret the results in a practical way.

As the Author states, wave energy spectra must be based on actual readings of seas in particular areas over a considerable period of time. The function of the Atlantic weatherships in this respect is well known, but could the Author say whether readings are being taken in many other places and, if not, whether the results already obtained are equally applicable to other oceans?

In addition to obtaining reliable data on ocean waves from a stationary measuring point, one must also consider the effect of the waves on the ship. Whilst the factors 1, 2, 4 and 5 on page 7 which influence the response amplitude operator can be evaluated, I should think that item 3 is not very amenable to calculation in a dry cargo ship or tanker where the longitudinal moment of inertia can vary within quite wide limits according to the distribution of cargo—unlike a passenger ship or naval vessel. Could the Author state whether these variations are likely to have much effect on the final results? The speed in storm conditions may also be considerably different from the normal service maximum.

In all this work one must not lose sight of the fact that the data being presented is for average storm conditions. Ships on unrestricted service must also be suitable for extreme conditions, and that these may well exceed the average by large amounts must be appreciated.

It can be seen from Fig. 14 that waves seldom exceed 50 ft. in height, but in extreme conditions twice this height is possible. The U.S.S. *Ramapo* after seven days of stormy weather in the Pacific in 1933, encountered a wave which, as a result of sighting from the bridge to a point on the mast, was reliably estimated to be 112 ft. high.

In the Southern Ocean storms can have a fetch which is greater than in any part of the world, but whether these areas have the same importance as in the days of sailing ships is doubtful.

I do not think that when the various seas of the world have been indexed and codified, we will see any great changes in design compared with ships as they are at present, but we will be in a better position to estimate the effect of changes in design on the stresses which the ship may be called on to withstand in service. Former practice in estimating longitudinal stresses was to use a wave of height directly proportional to the ship's length, and to vary the allowable total stress with length of ship to take account of the reduced probability of the larger ships meeting waves equal to their own length. We now use in L.R. a wave proportional to the square root of the ship's length but still with a variation in stress with length. It seems desirable and probable that in the future we will be using a constant stress regardless of the length of ship, but the standard wave for each length will vary, not only with the length of ship, but also with the form and speed. This would seem to be a better concept than the present one. The allowable stresses could be related to the average conditions for which data had been obtained. To take account of extreme conditions for which data would also be available, the allowable stress for such conditions could be close to the yield point of the material. Regarding these extreme conditions one would need to know for various heights of waves the corresponding maximum length of single waves.

In conclusion, I should like to ask what is the purpose of the dotted diagonal lines in Fig. 2, and whether the letters a1, a2 and b1 are related to the formula for this sea condition.

AUTHOR'S REPLY

Before referring to the contributions to the discussion, it is perhaps appropriate to reiterate that the whole subject is still in a very fluid condition. Since the paper was written there have been additional suggestions on the proper way to interpret information, both long- and short-term, gained from strain gauges fitted to ships in service and the application of the theory of probability to this work will no doubt continue to advance. The same remarks apply also to the theoretical approach to the matter by way of the energy spectrum, and Mr. McCallum's remarks on the merits of the various sea spectra emphasise this point of view.

Mr. Boyd raises the difficult question of the relative importance of the number of reversals sustained and amplitude of these reversals. The Author is of the opinion that fatigue in the classic sense of the word is not a factor of importance when considering the strength of ships, and therefore the approach of selecting an acceptable probability in relation to amplitude is the proper basis to work on. Mr. Boyd's query regarding the use of the response amplitude operator demonstrates that the paper is not as clear in this respect as it should be, and some further explanations of the table on page 7 might be appropriate.

It will be seen that the R.A.O's (in inch pounds/inch) appropriate to various ratios of $\frac{\lambda}{L}$ are given in the third column, and in the fifth column, the appropriate frequency of encounter. In the sixth column there is the $Fn(\omega)$ that is, the wave spectrum value appropriate to the frequency of encounter, derived from Fig. 13. The B.M.^{2} also appropriate to the specific ratio of $\frac{\lambda}{L}$ is found by multiplying the R.A.O.^{2} by $Fr(\omega)$, and the bending moment spectrum is constructed in this way. Fig. 2 merely shows how the wave form is altered by the addition of three simple waves,

and Fig. 3 shows that the spectrum of these waves is a discrete and not a continuous spectrum.

Mr. Davies, by pointing out that there is little likelihood that the application of these new methods will lead to any immediate reduction in scantlings or redistribution of material, has answered several questions, including that raised by Mr. Hobson. It is most unlikely that there will be any departure in either general principle or in detail from present standards because of the impact of the new methods for determining longitudinal strength until a great deal more experience and knowledge has been gathered. This applies to the point raised by Mr. Beken. It will be a long time before N_1 will be modified by N_2 though there is a theoretical justification for doing so.

Several of the contributors have called attention to the confusion of the storm sea and the chance of waves of extreme height being encountered. Mr. Clemmetsen and Mr. Lockhart in particular do so, and this phenomenon of course is the basic difficulty about probability theories. There is always a wave height greater than that defined at any level of probability.

Mr. Clemmetsen points out that several of the factors entering into the determination of R.A.O.'s are very difficult to estimate, and the effect of speed of ship is in that category at present. As is stated in the paper, these effects can be determined mathematically but the work is tedious, complicated and to some degree the results are uncertain.

Mr. Lockhart refers to the U.S. experiments on T-2 tanker models, and it is through experiments of this kind that knowledge of the behaviour of ships in regular and irregular seas is obtained. He is quite correct in thinking that the experiments have been carried out with a model and a few strictly limited conditions; it is necessary to reduce the number of variables in these experiments as much as possible but in the future no doubt the experiments will be extended to cover ballast and other conditions.

Mr. de Wilde, in his remarks on the relation of a standard wave to the Neumann spectrum, touches a most important point. His remarks emphasise the difficulty of reconciling the ideal concept of a constant stress with the sea spectrum described by the Neumann approach and this is only one of the difficulties which have to be resolved sooner or later.

Mr. Brinton may be reassured that in Lloyd's Register strength calculations are still done in the classic way and the Author is quite sure that this obtains throughout the shipbuilding world. There will be, undoubtedly, changes made as the detailed applications of new theories become possible, but the Author thinks that it will be a number of years before this is done. This answers also Mr. Hobson's question.

In conclusion, the Author would like to thank those who took part in the discussion, for their useful and interesting comments. He hopes he has clarified at least a few of the obscurities in this very difficult subject.

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INTERESTING INVESTIGATIONS

by

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Interesting Investigations

By J. H. Milton

The object of this short paper is to endeavour to present something of interest to marine engineers in general: it deals entirely with troubles which have occurred, been investigated and could repeat themselves, although not, it is hoped, on the same vessels!

The necessity for higher vessel speeds with economy has resulted in the squeezing of more power into smaller spaces on board ships; and this has been achieved by the use of higher temperatures and pressures, fabricated instead of cast structures, gearing of harder materials and higher tooth loading, etc., all of which have presented fresh problems.

Generally speaking, the comparatively smaller high efficiency machinery of the present day is more intricate and needs more skilled attention than the older types of propelling units, which appeared, at any rate, to lead a much more leisurely life. Compare, for instance, the fascination of a large quadruple steam reciprocating engine with the noise and bustle of blown geared diesels.

It is in troubles which have occurred that we are interested to-night, but before describing these, two points are worthy of consideration. Firstly, when undertaking any investigation, normal causes of the trouble in question should be thoroughly probed before embarking on more intricate and involved lines of action. It is surprising how often weeks of highly technical work are put into an investigation, which is subsequently proved by a roundabout process to have a simple solution. A typical example of the foregoing is the first investigation detailed under the heading of "Steam Reciprocating Engines".

Secondly, in some cases investigations are effected and information obtained, but because no previous research has been done on the subject in question, no just comparison can be made, and therefore no logical conclusion arrived at. This brings us to the difference between research and investigation: when doing the latter one is often driven to thinking that, if only some research work had been done beforehand, the investigation would not have been necessary.

At the risk of contradiction, one could include in such researches: main engine bedplate deflections in a seaway, the cause and nature of hydrogen fires in boilers, the cause of excessive after end noise from some propellers, oil engine exhaust pulse vibration troubles, after end hull vibration resulting from unsuitable after body/propeller combination, etc., etc.

As in my paper on "Machinery Breakdowns", I propose to deal with the subject in categories, namely:—

- Oil Engines;
- Steam Reciprocating Engines;
- Turbines;
- Boilers;
- Shafting and propellers;
- Vibration.

Investigations of troubles experienced in each category, and wherever possible the recommendations made for their rectification, will be described.

OIL ENGINES

Bedplate Deflections in a Seaway

It was suggested at the beginning of this paper that there were several lines of research requiring attention, one of these being bedplate deflections in a seaway. Such an investigation was requested by a shipowner who had several vessels building with the same type of 2SCSA machinery. In the case of one vessel, the main engine fabricated bedplate scantlings were somewhat lighter than in the others, and the possible deflections of this lighter bedplate under working conditions in a seaway were queried.

Examination of the plans of this bedplate showed that inside it and on each side of the sump there were lightening holes in each of the cross members for the length of the engine. It was decided therefore to stretch a steel strip about 2 in. wide by $\frac{1}{8}$ in. thickness through these holes, to anchor it at one end and to tension it by screw tightening gear at the other end—the tension to be kept constant by referring to strain meter readings taken from a strain gauge bonded to the strip. At the lightening holes through which the strip was running, "Desynn" gauges (in reality sensitive potentiometers capable of operating through about $\frac{1}{2}$ in. movement) were mounted, the actuating pins of which engaged in small saddles bolted on to the steel strip (*see* Fig. 1).

The leads from the "Desynn" gauges were fed into a galvanometer recorder and prior to the actual test each "Desynn" gauge was calibrated by using feeler gauges between the gauge actuating pins and the steel saddle blocks, thus establishing the magnitude of the galvanometer recorder movement for each gauge. It was feared that vibration of the steel strip might occur under working conditions and upset the result, and provision was made to counter this eventuality by

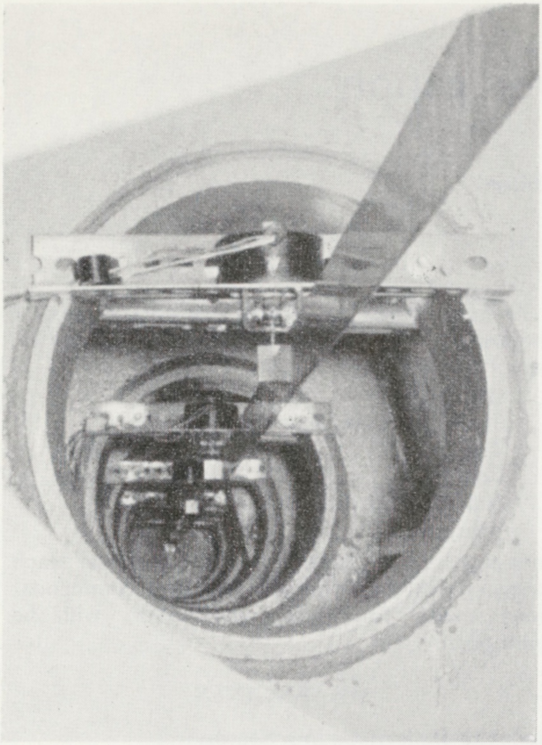


FIG. 1

attaching a plate to the strip submerged in an oil reservoir to act as a dashpot-type damper.

Vibration of the strip did not occur, the apparatus operated quite satisfactorily and has been used on two different types of oil engines. Fig. 2 shows the arrangements as fitted to a Doxford engine, in which case, as can be seen, it was used through the "A" frames and not through the bedplate.

Fig. 3 is a typical record showing the movements recorded at sea in a seaway.

It is worthy of mention at this point that an individual investigation of movement in a particular engine is one thing, but before any opinion can be formed on the implication of such movement, a comprehensive research programme embracing measurements taken statically under all different types of loading and ballasting, also dynamically in different sea conditions, would have to be undertaken.

Crankshaft Investigations

Much can be written on this subject, and it is only proposed to deal briefly with the framework of such investigations and subsidiary points of interest.

When a crankshaft in an oil engine, of a well tried design, suddenly fails after only a short service, it is always due either to unusual operating conditions or an original defect in the shaft.

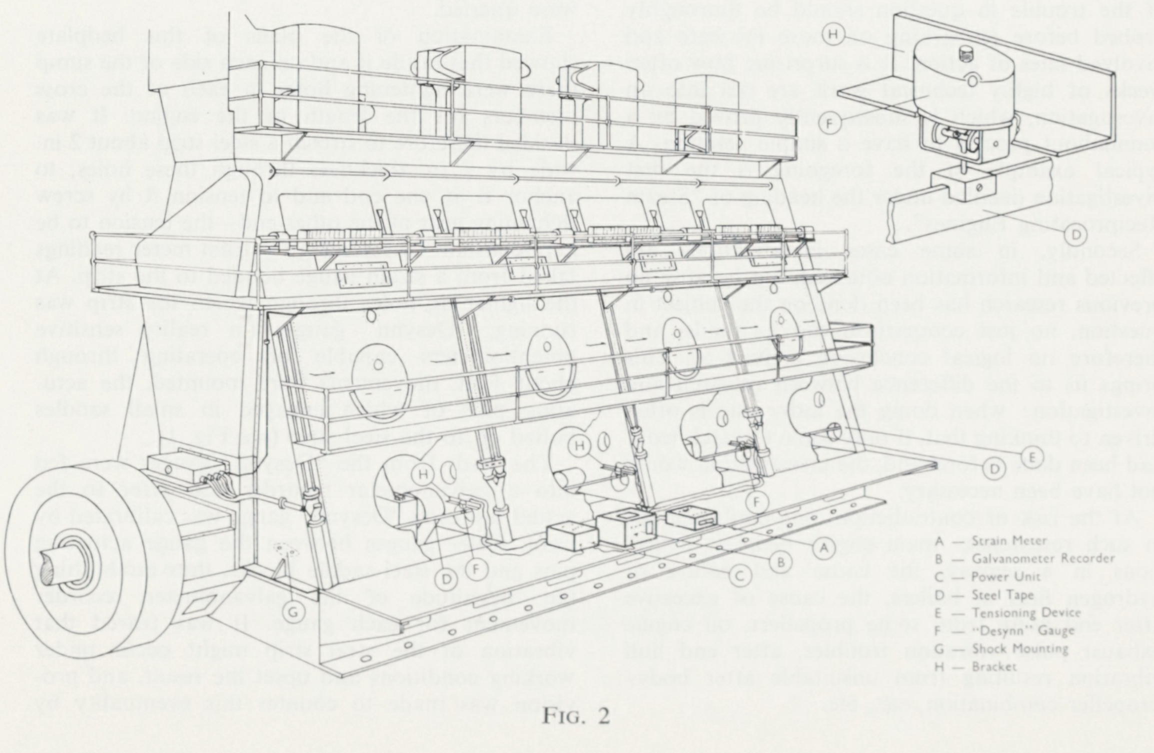


FIG. 2

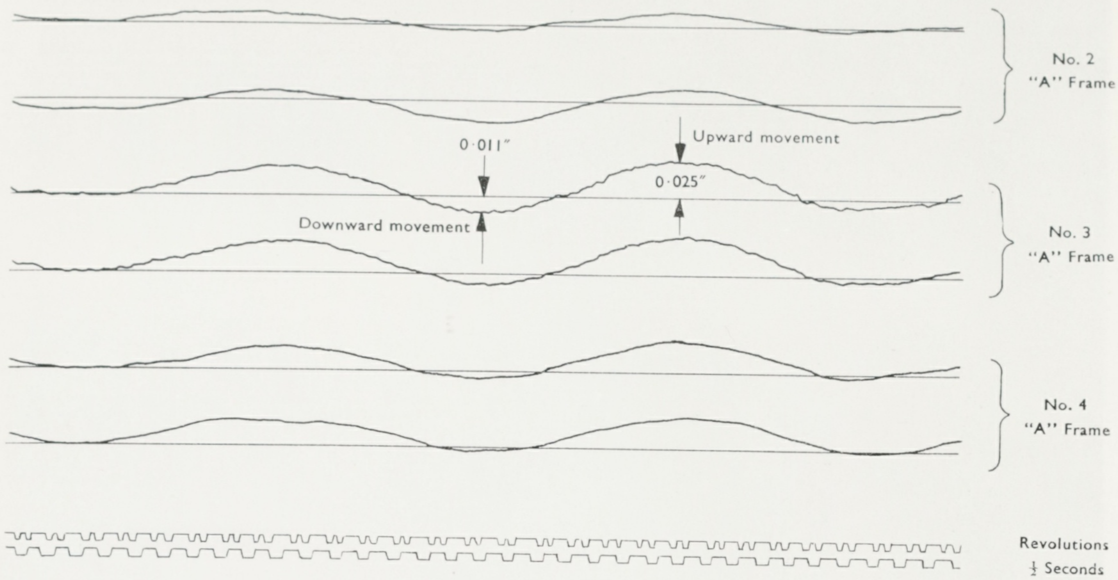


FIG. 3

Under the category of unusual operating conditions can be included:—

1. Misalignment;
2. Torsional and axial vibratory stresses;
3. The effect of hydraulic loading in cylinders;

and under original defects:—

1. Latent defects in the forgings or castings with which the shaft is built;
2. Shrink fit discrepancies;
3. Stress concentrations at machining discrepancies.

When investigating a failure, the first step is to examine the fracture; this, to one experienced in such matters, can by virtue of its direction and nature of the fractured surfaces (if these are visible) present valuable evidence as to the cause of failure. Subsequently, provided the shaft has not parted completely and caused serious consequential damage, it is usual to endeavour to ascertain the alignment conditions under which the crankshaft has been operating (this normally being effected with the Taylor Hobson alignment telescope or taut wire equipment). At the same time discussions are held with the operating personnel and the log books, etc., are consulted with a view to finding out whether any unusual operating conditions have been experienced which could have had an influence on the failure.

Assuming that the alignment is satisfactory, no unusual operating conditions have apparently occurred and an examination of the bedplate and chocking shows same to be in efficient condition, metallurgical examination of the break is effected as soon as specimens can be cut from the shaft. This examination, as mentioned previously, nearly always provides valuable evidence inasmuch as it may indicate that the failure has been due to torsional or bending stresses, a faulty forging or a casting defect. The characteristics of such defects are shown in Fig. 4 (a, b, c and d).

As a matter of interest, and assuming a shaft has been shown by metallurgical examination of the fracture to have failed through torsional or bending stresses, it is proposed to describe briefly how the magnitudes of such stresses have been measured under operating conditions.

Where, as in the case of Doxford engines and some others, the crankshaft oilways are continuous from one end to the other, electrical cables have been passed through and connections made to strain gauges on almost all positions on the crankshaft. In the case of some of the smaller engines where diagonal drilling from individual journals to crank pins is used for oilways, it is usually impossible to fix strain gauges at any position other than the aftermost journal and pin.

Assuming the connections can be made to the strain gauge position, it is usually necessary (except in the case of crankshafts with undercut fillets) to fit, temporarily, special bearings having a part of the white metal champhered away to clear the strain gauges. Fig. 5 shows the method of installation of the gauges.

Connections to these gauges are made via electric cables led through the oilways to slip rings on the end of the crankshaft, brush gear connected to either a Kelvin Hughes 4-channel recorder or an ultra-violet recorder being used for measuring and recording the strains occurring.

A typical record showing vibratory strain recorded by this latter instrument is shown in Fig. 6. The strain gauges employed in this technique are suitably orientated and connected so that either bending or torsional strains are recorded.

From the strain recorded at any point the stress can be calculated, although it must be borne in mind that this is the stress locally in way of the particular gauge position. A slight alteration in the position of a gauge, for instance further round

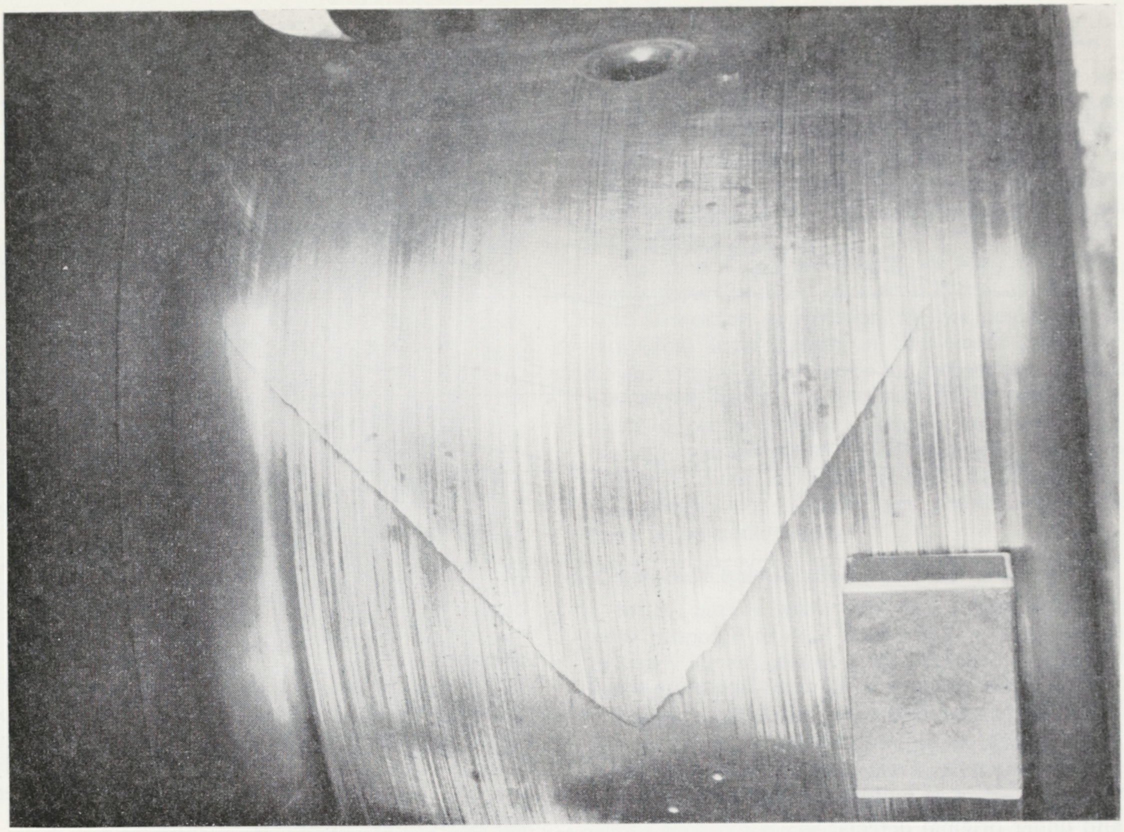


FIG. 4(a) Typical torsional failure

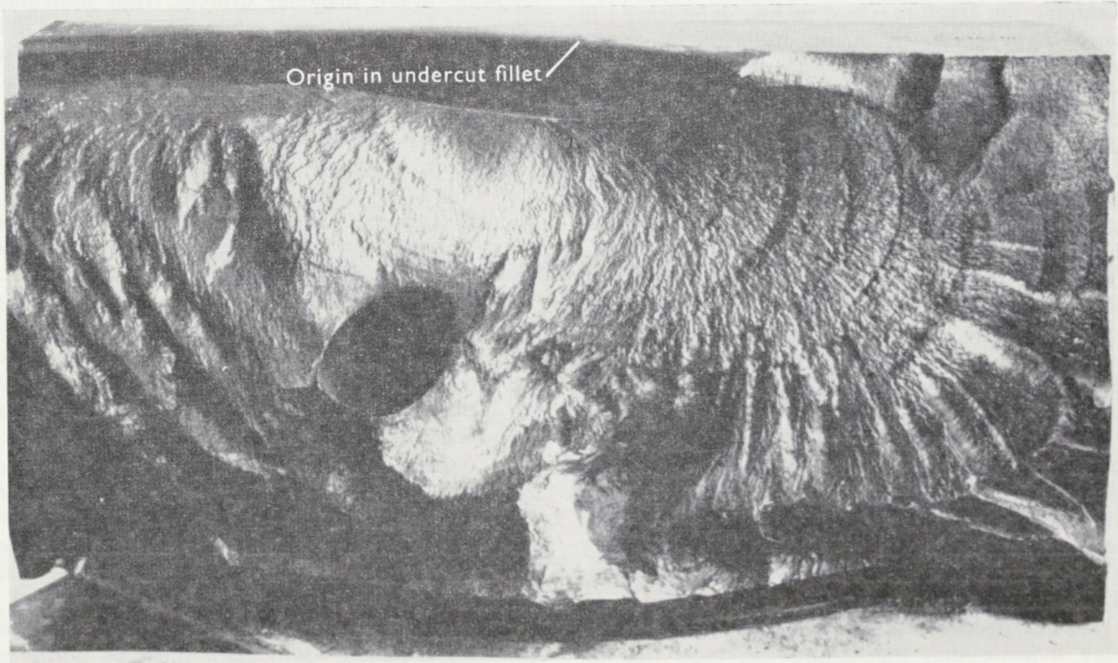


FIG. 4(b) Typical bending fatigue failure

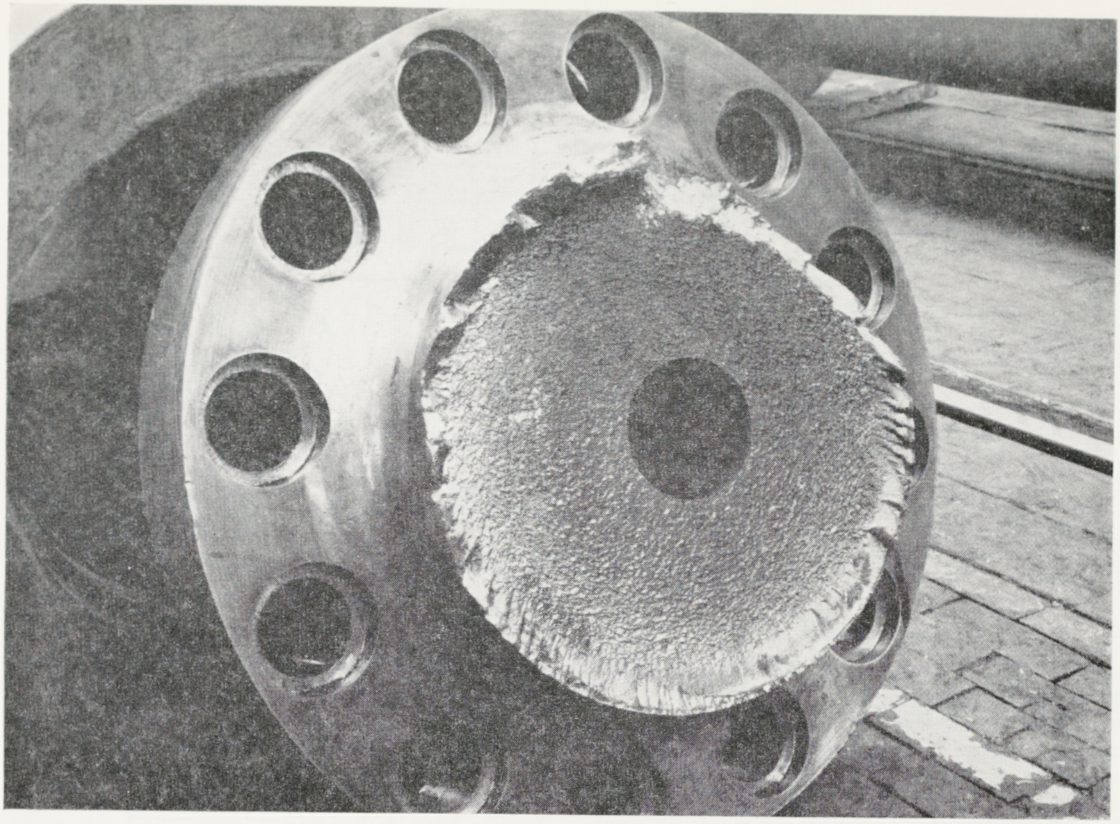


FIG. 4(c) Journal failure through faulty forging

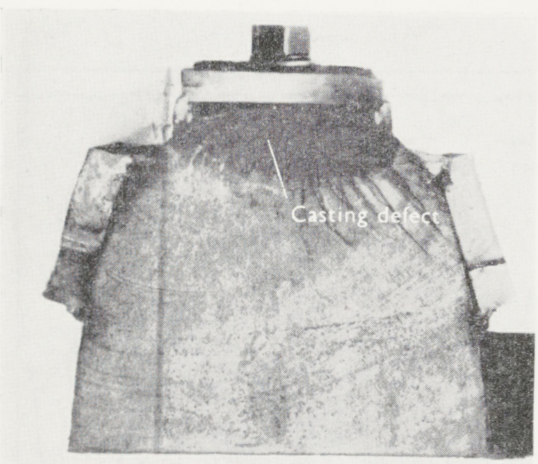


FIG. 4(d) Crank web failure through faulty casting

a fillet, may give a very different result: in other words a group of gauges can pinpoint a stress concentration, although the base lengths of the strain gauges must be considered when interpreting the result.

In the case of crankshafts with different forms of crankpin fillets it is a relatively simple matter to load up a working piston statically either mechanically or hydraulically, with the engine on TDC, and to ascertain by the application of strain gauges along the underside of the crankpin the stresses in existence along the pin and in the fillets, thus obtaining stress concentration factors for different fillet forms.

In such investigation work on components which have failed it is usually possible on a similar component, in service, to ascertain the working stresses in way of the failure. The main difficulty arises, however, when, bearing in mind their complex shapes, an endeavour is made to assess a safe working stress for any one part.

Bearing Distortion

Whilst investigating failures of different types of plain bearings, opportunity has been taken on many occasions to ascertain whether the bearings in question distort on tightening up the bearing bolts. It has always seemed incredible to me that so much time, care and hard work are expended in bedding white metal bearings on to journals and pins whilst in a free condition, when often the

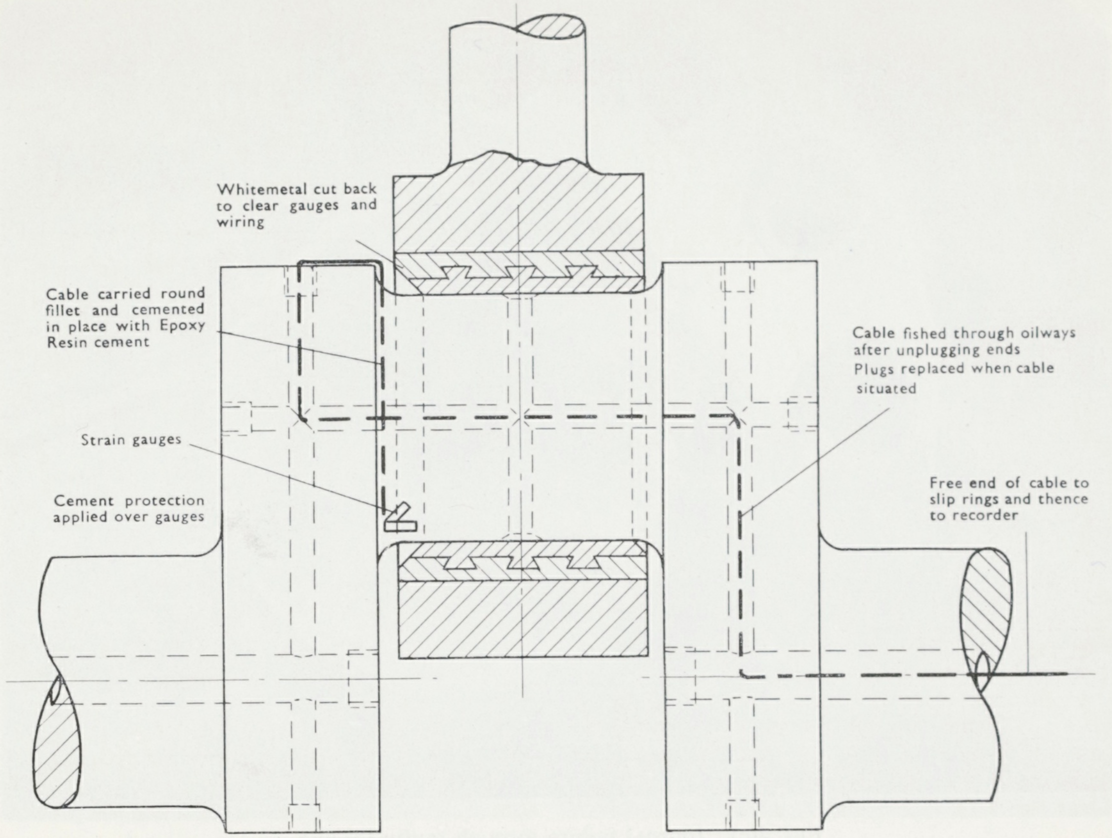


FIG. 5 Simplified diagram of strain gauge installation on crankpin

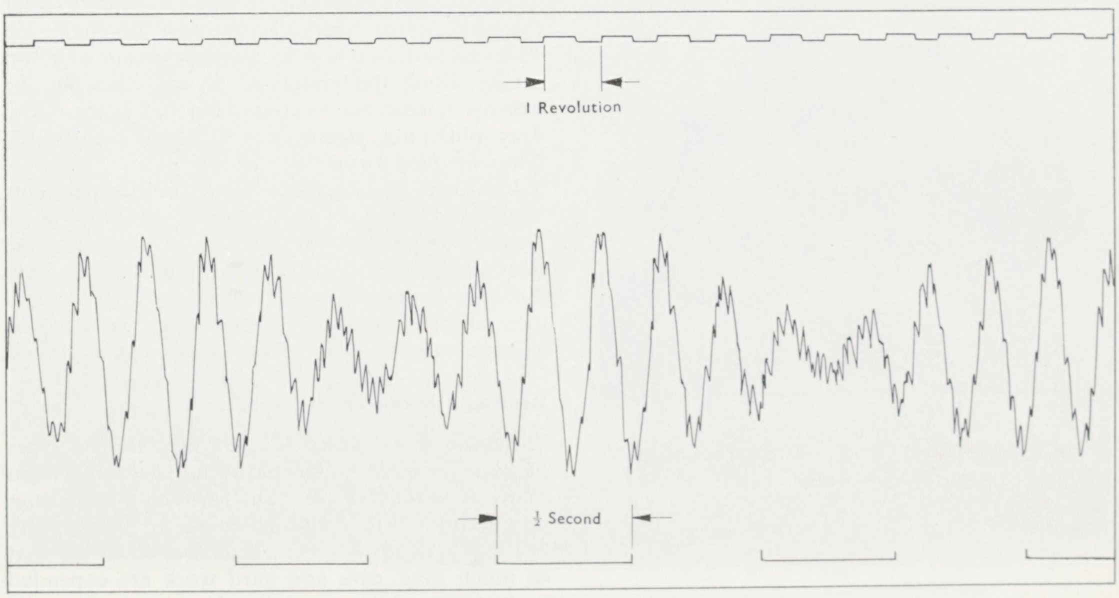


FIG. 6

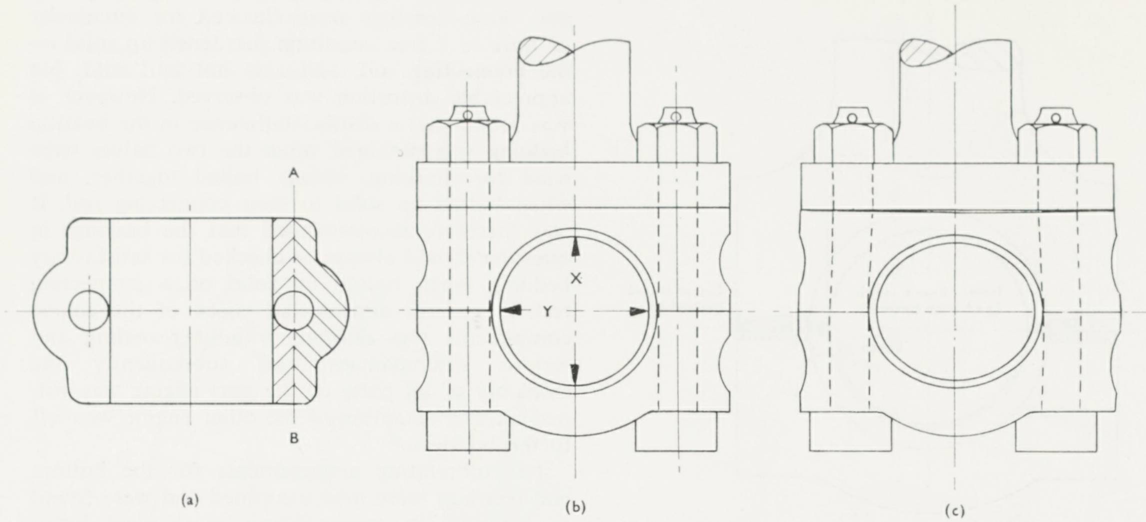


Fig. 7

bearings alter their shape appreciably and hence destroy this bedding, on bolting up.

In one case dealt with recently a bearing for an 11 in. diameter journal decreased its crown to base diameter by ·007 in., and increased its diameter in line with the jointing face by ·002 in., when the bolts were pulled up to their correct stretch.

Examination of the cross section of a pair of bearings at the jointing surface gives, I think, some indication as to whether they will distort on bolting up.

To avoid dynamic bending stresses in the bolts it is usual to position the bolts as near in to the journal as possible (*see* Fig. 7(a)). In achieving this object, however, the balance of jointing face area either side of the centreline A–B drawn through the bolt hole is badly upset, so that on tightening the bolts the area nearest the bearing bore compresses slightly more than the outer area, with the result that the diameter "X" decreases and "Y" increases (*see* Fig. 7 (b and c)).

This effect is in some cases aggravated still further by the use of so-called "horse-shoe" liners between the jointing faces (*see* Fig. 8 (a)). These liners, often cut by hand and purposely left short at "C" to avoid touching the journal, further upset the balance of area about the centre-line "A–B".

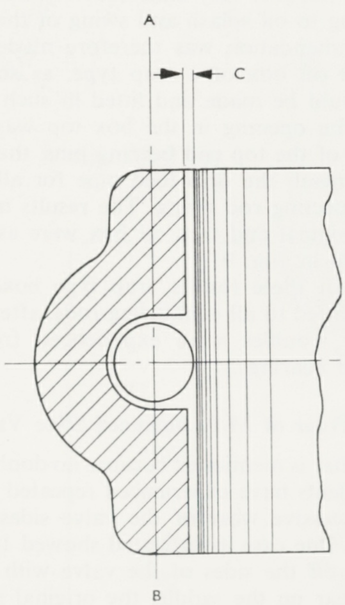


Fig. 8(a)

When investigating failure of bearings, be they bottom ends, top ends, eccentric straps or even main bearings, it is well, if possible, to check the bore diametrically in, say, three positions with the bolts hand tight, and then re-check with the bolts hardened up to the proper stretch.

Main bearings were mentioned as, in the case of one steam reciprocating engined installation, the stiffness of the bearing keeps was inadequate and, on hardening up the main bearing nuts, the keeps bent slightly and closed the bearings in on their horns.

I think it well to digress here and state that whilst investigating top end troubles, various ways of adjusting spherical bottom end bearings were encountered. In one of these the liners in the inner bearing were adjusted until correct crankpin leads were obtained, simultaneously, leads were put between the liners of the outer bearing, and the bolts hardened up each time the double set of leads was taken. On obtaining the correct inner leads, the thickness of the leads between the outer liners was measured, and a liner equivalent to this plus the outer bearing clearance was inserted. In this method hardening up the bottom end bolts with lead wire between the outer liners whilst making adjustments is, I think, inviting distortion to take place.

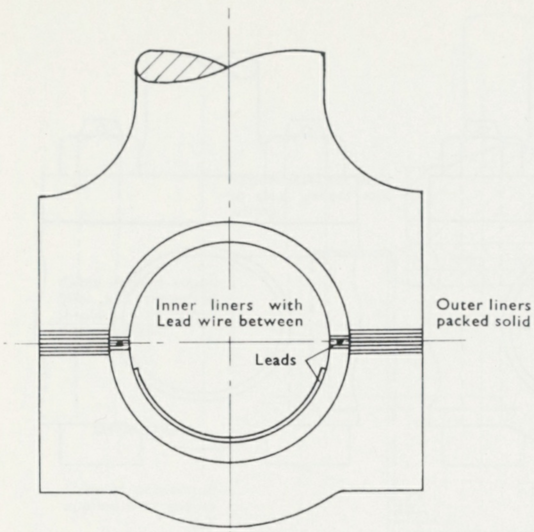


FIG. 8(b)

If inner and outer bearings are to be adjusted simultaneously, the correct method is to adjust on the outer liners until a crankpin lead thickness corresponding to the total clearance, i.e., inner plus outer bearing, is obtained, and simultaneously to have a lead between the liners of the *inner* bearing (*see* Fig. 8(b)). So that when the lead from the inner liner is measured, a liner of this thickness minus the spherical clearance desired can be inserted in the inner bearing. In this method the bottom end bolts can be solidly hardened up at each adjustment without any fear of distortion taking place.

STEAM RECIPROCATING ENGINES

Bearing Troubles

This type of machinery, at one time extensively used as the propulsion unit of practically all types of vessels, is still specified for certain specialised craft, such as dredgers, etc., and it was on one such craft that the following investigation was effected.

For nearly a year the builders of a twin screw dredger propelled by two triple expansion steam engines having cylinders 350, 535 and 880 mm. diameter by 460 mm. stroke, had experienced what seemed to be endless trouble with overheating of bottom end bearings. The clients were pressing for delivery and an investigation was requested.

The possibility of torsional vibration affecting the engine performance was first investigated and no untoward features were revealed. The crankshafts of both engines were removed from their bearings, checked for truth, parallelism and surface finish, and after some slight lapping of pins and journals, were considered satisfactory. The alignment of all main bearings was then checked by mandrel, and after bedding each crankshaft, the alignment of this to its thrust shaft was checked and found in order. The alignment of connecting rods to crank pins and journals, etc., was also verified and found good.

Bottom end bearing distortion was suspected and these bearings were checked for circularity of bore in a free condition, hardened up solid on the connecting rod, and also hot and cold. No appreciable distortion was observed. However, it was found that a distinct difference in the bearing bedding was obtained when the two halves were tried for marking, simply bolted together, and when bolted up solid to their connecting rod. It was therefore recommended that the bearings in question should always be checked for satisfactory bedding when bolted up solid on a connecting rod. A general dimensional check of the engine components was effected without revealing any serious discrepancies and subsequently the assembly of all parts of the port engine was witnessed. The assembly of the other engine was left to the builders.

The lubricating arrangements for the bottom end bearings were next examined and were found to embody an engine-driven mechanical lubricator, feeding an oil box on top of the top end bearings, this oil box having a pipe down one side of the connecting rod to the bottom end bearing.

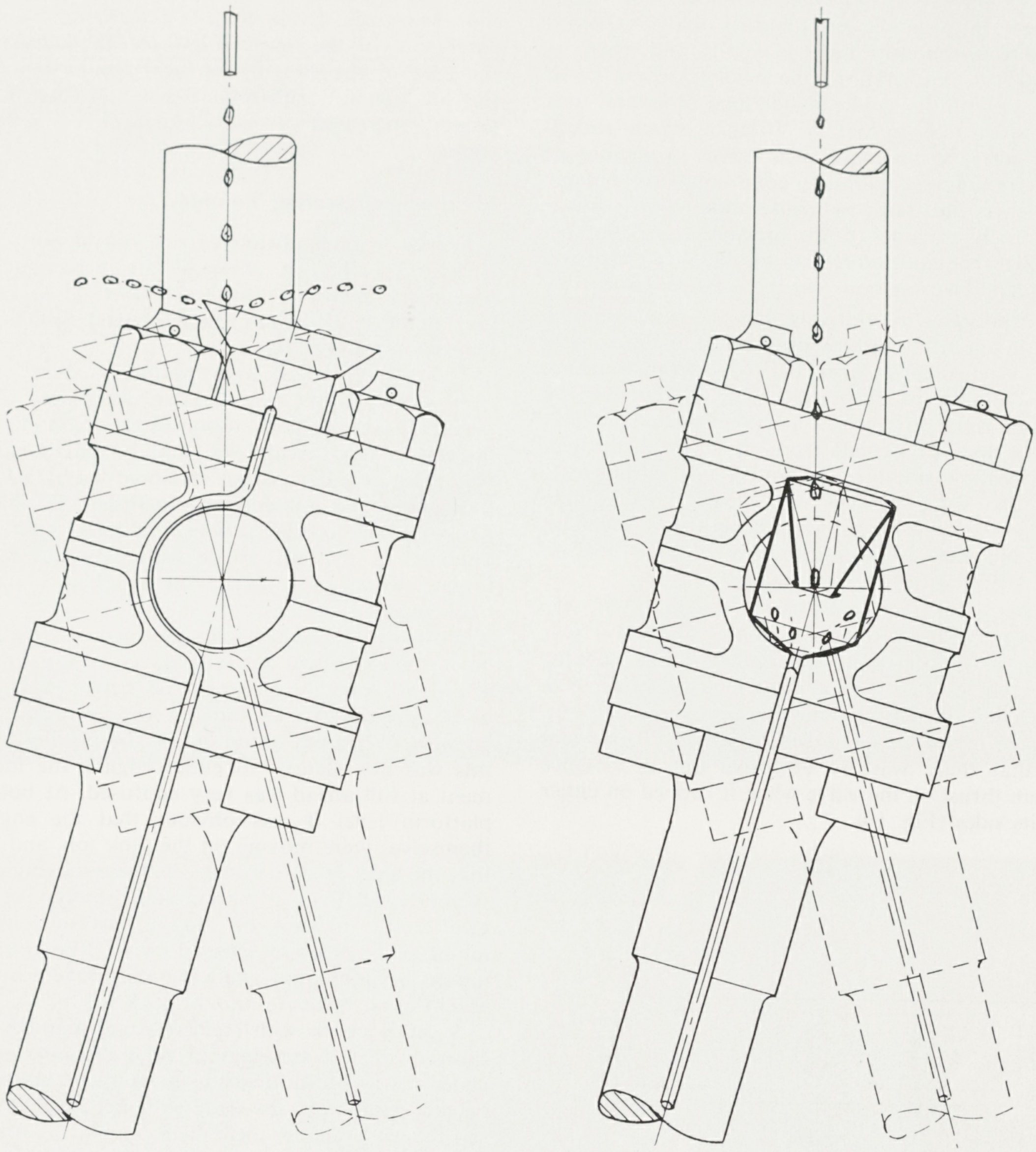
The bottom end bearing troubles occurred at the higher r.p.m. between 150 and 180, and as this range is high for drip feed lubrication, a request was made that at the next engine trial one of the connecting rods should be fitted with two sets of bottom end bearing lubricating gear, i.e., the normal one and another leading down the other side of the connecting rod to a small vented tank, bolted to the connecting rod foot. The small tank was fitted with a screwed plug for draining into a measuring glass. At the engine trial, measurements were made of oil reaching the tank during five-minute runs at different r.p.m., and it was found that the oil box and pipe arrangement, as fitted on top of the top end bearings, was satisfactory up to 130 r.p.m., but was very unsatisfactory above that speed owing to oil splash and swing of the box.

A recommendation was therefore made that a re-designed oil box of a trap type, as shown in Fig. 9, should be made and fitted in such a position that the opening in the box top was at the centre line of the top end bearing pins, thus being directly beneath the drip feed pipe for all angles of the connecting rod swing. The results obtained with the original and trap oil box were as shown in the table in Fig. 9.

In view of these results, trap type boxes were made and fitted to all connecting rods, after which no further troubles were experienced from the bottom end bearings.

Excessive Wear of M.P. Balanced Slide Valves

This matter is mentioned because no doubt many superintendents have experienced repeated trouble through excessive wear of the valve sides within the saddle. One case investigated showed $1\frac{1}{4}$ in. to have worn off the sides of the valve with corresponding wear on the saddle, the original rubbing strips at the sides having worn away completely so that the whole area of the valve sides could



Original oil box

Re-designed oil box

R.P.M.	Oil delivered by mechanical lubricator during 5 mins.	Oil reaching foot of connecting rod from original oil box	Oil reaching foot of connecting rod from re-designed oil box
70	10 ml.	10 ml.	9·2 ml.
100	15 ml.	12·8 ml.	15 ml.
120	17 ml.	17 ml.	13·5 ml.
160	20 ml.	6·2 ml.	20 ml.
180	20 ml.	4 ml.	15·5 ml.

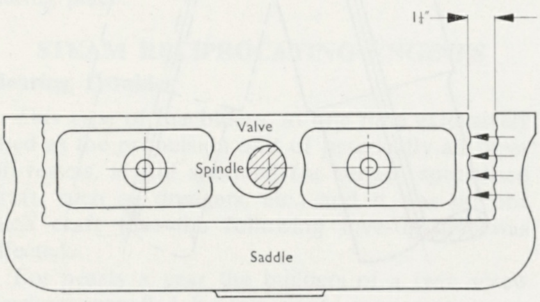
FIG. 9

contact the saddle. So-called balanced slide valves depend in principle on the fact that the steam loading on opposite faces is equal ; and when, as in the case in question, one whole side can be in contact with the saddle, a side load of several tons can exist due to out of balance steam thrust. Obviously, to maintain such valves in a more or less frictionless operating condition, any outside influences tending to cause side wear—which ultimately excludes steam pressure from the surfaces in contact and upsets the balance—must be avoided. Two methods can be used to do this: —

(a) The valve is positively located on the spindle in the athwartships direction and the spindle is relied on to keep the valve from touching the sides of the saddle. The spindle must be of rigid proportions and well guided, so that it does not deviate from its true path through possible side thrust from the link motion.

(b) The valve is kept well clear of its spindle (from which outside influences can originate) and has a minimum area of side bearing strips, with fine clearances.

In the case under discussion method (b) was adopted with satisfactory results. The valve was given $\frac{1}{4}$ in. all round clearance on the spindle. Two out of the original four bearing strips were machined off entirely, and the remaining two had their effective area reduced by about 30 per cent so that there was no excessive out of balance steam thrust on the valve when it rubbed on either of its sides (Fig. 10).



Valve and saddle as found

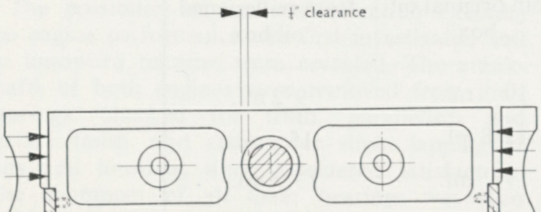


FIG. 10 Valve and saddle as repaired

Perhaps the foregoing is elementary and common knowlege. Even so, it is interesting, as it is surely one of the few cases where an increase in the area of the wearing surfaces would result in the loading and subsequent general wear being greatly increased if these surfaces come into contact.

Main Engine Seating Troubles

Survey work is always carried out in port, and it is impossible to envisage the behaviour of machinery under working conditions at sea, from the condition of a few parts opened up on the continuous survey basis.

The owners of a large passenger vessel with twin screw steam reciprocating machinery were perturbed about the vibration of the main engines under working conditions, and an investigation was requested. The vessel was boarded at a U.K. port and sailed immediately. In order that a preliminary idea of the problem in hand could be obtained, a visit was made straight away to the Engine Room to observe the machinery under manœuvring conditions.

Most steam reciprocating engined installations have some vertical movement, especially at the after end, but these two large triple expansion units appeared to be moving in every direction possible. When one stood on the top grating, since this was not secured to either engine, the movement at full ahead was very confused. At bottom platform level it was obvious that the engines themselves were moving on the tank top, and also that the axes of the crankshafts were moving considerably relative to surrounding objects—at the after ends, in view of the various movements taking place, it was impossible without instrumentation to differentiate between the movements of crankshafts, bedplate and/or tank top.

A quick check with feelers during manœuvring showed the L.P. bearings to have excessive clearances—30 to 50 thousandths of an inch ; enquiries on this point were answered to the effect that they ran hot with smaller clearances.

A programme of investigation for the sea passage was then carried out. All accessible main bearing clearances were checked, the power balance of the engines was obtained, vertical deflections of the bedplate under working conditions were measured, the vertical movement of a bedplate relative to the ship's structure was ascertained, the holding down bolts and double bottom tanks in way of the machinery were examined whilst the machinery was developing normal full power, the movements of main engines and shafting were recorded by vibrograph, etc.

It had been the practice always to run with the engine room double bottom tanks full, as this, it was stated, reduced the general vibration ; on request they were pumped out and the doors removed. The internal examination of these tanks at sea, made by the light of a torch, with loose water swishing about and the investigators clad only in rubber boots and boiler suits, was quite an experience. The noise of the main engines

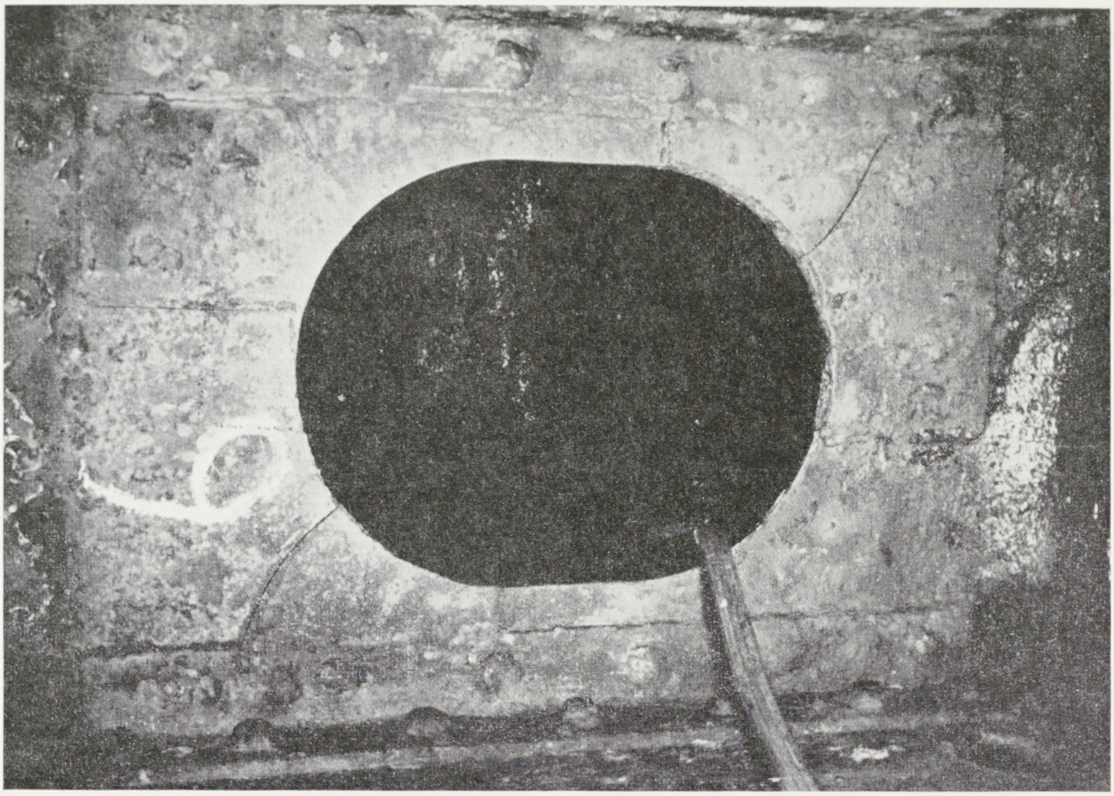


FIG. 11

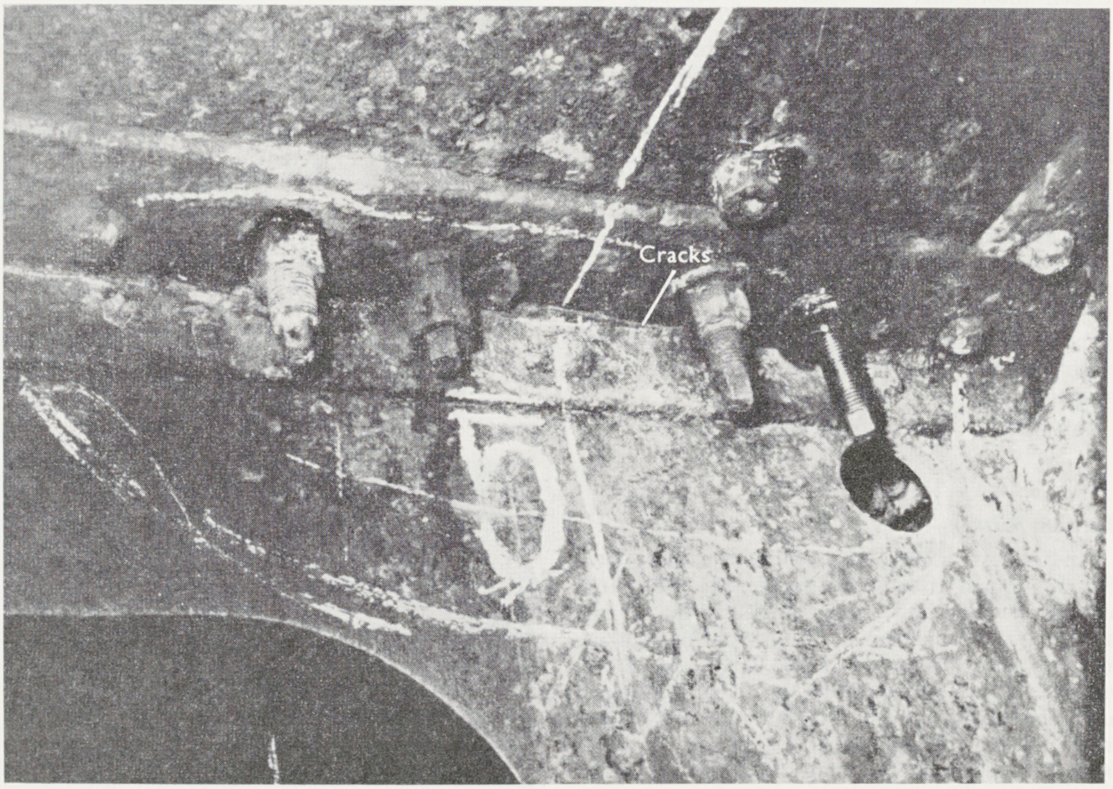


FIG. 12

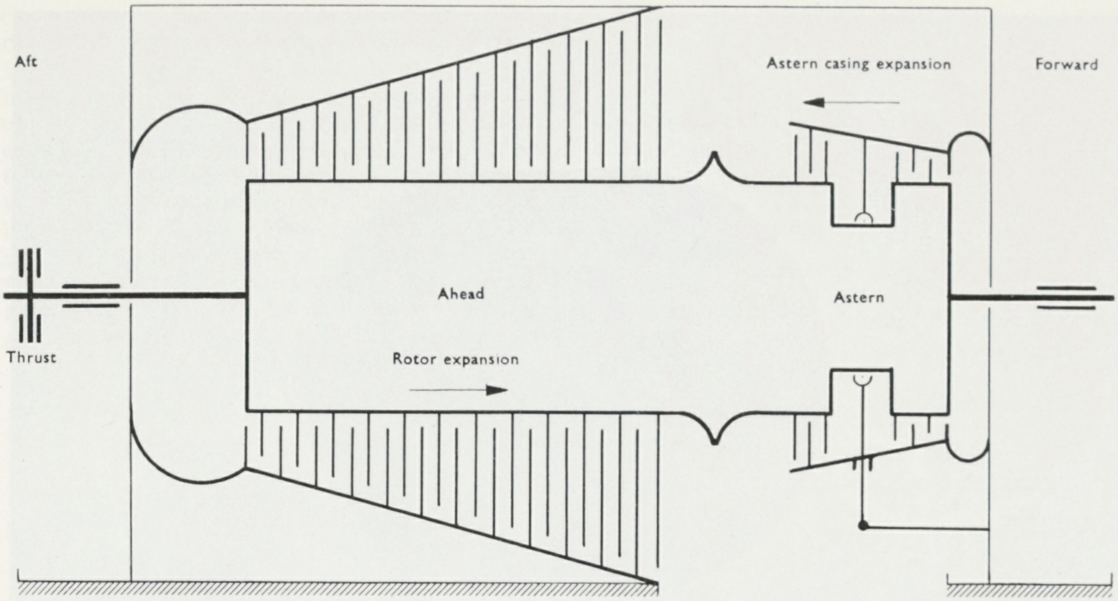


FIG. 13 Diagrammatic arrangement of L.P. turbine

thumping up and down on the tank top and the sight of holding down bolts moving up and down, cracks from intercostal lightening holes and in top angles opening and shutting, top angles moving on their rivets, etc., were very enlightening if not alarming, and would have been even more so had it been known that the vessel was proceeding full ahead in fog!

Typical examples of the nature of the fractures, which were numerous, are shown in Figs. 11 and 12.

Examination in port of the bedplate chocking showed this to be in very poor condition, it being impossible in many cases, after removing the holding down bolts, to prise out the chocks through severe indentation and erosion.

As would be expected, the poor condition of the chocking was again evident when the crank and thrust shaft alignment was checked, the net result being that ultimately both main engines were removed for seating repairs, re-alignment and re-chocking to be effected.

TURBINES

Turbine Seizure

An interesting case of turbine machinery (*see* Fig. 13), which after several short runs ahead and astern jammed until the pressure of the ahead nozzles was brought up to 18 kgs./cm.^{2}, occurred in a new vessel a few years ago. The ship was in restricted waters and once the turbines had moved, an order for full astern and several ahead movements was effected until a heavy vibration occurred which necessitated an immediate stop being made. The turning gear subsequently failed to turn the machinery until the L.P. turbine was disconnected, and on opening up this turbine the astern blading was found to be extensively damaged. All four rows of astern impulse blading were found to have made contact with their adjacent stator blades or nozzle plates (*see* Figs. 14 and 15), there being no evidence of radial rubbing of the rotor at any of the glands or labyrinth seals.

In this case it was considered highly probable that the seizure occurred through the relative expansion of rotor and casing taking up all the axial blading clearance at the astern end. It will be noted from the diagrammatic arrangement of this turbine (Fig. 13) that the thrust is at the after end and, such being the case, the axial blade clearances at the astern end could have been taken up through:—

(a) The astern manœuvring valve not being properly shut whilst running ahead—this resulting in the astern turbine performing work on superheated steam with ensuing high temperatures.

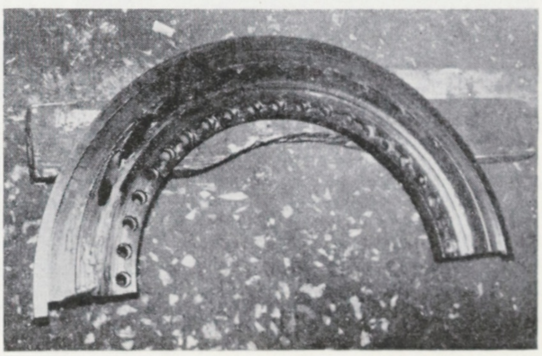


FIG. 14 Astern inlet nozzle plate showing rubbing

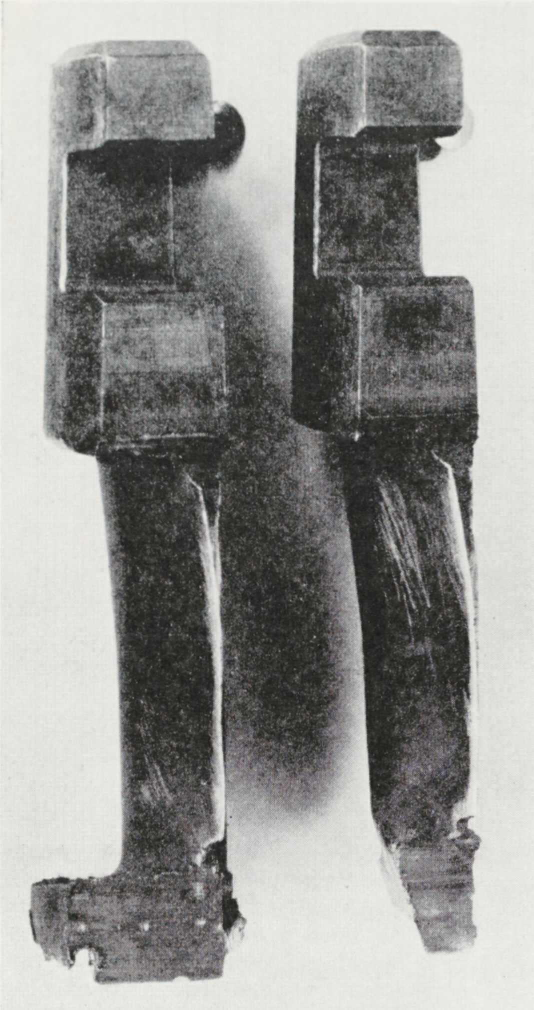


FIG. 15 Rubbed forward face of astern blading

(b) Running astern with insufficient vacuum, this resulting in overheating of the ahead end of the turbine.

Under these conditions the expansion of ahead and astern parts of the rotor is forward, i.e., away from the thrust, whereas the astern casing, being anchored only at the forward end, will expand aft.

The axial clearance of the astern blading in this case was increased by 100 per cent and as far as is known there has been no recurrence of the trouble.

The Measurement of Distortion of Turbine Casings

A method of measuring the deflection of oil engine bedplates under working conditions in a seaway has already been referred to. "Desynn" gauges feeding into a twelve-channel recorder were used. In the case of relatively small movements inductive proximity pick-ups have proved very useful as they can be used to record variations in an air gap, in other words no actual contact is required. These gauges were used recently to measure the hog and sag of an h.p. turbine casing under normal operating and manœuvring conditions. A steel tube was arranged along the side of the turbine at joint level ; it was firmly fixed to the casing at one end and allowance was made for differential expansion at the other by a sliding connection. A rigid clip attached to the tube at mid-length of the turbine carried two differential inductive proximity pick-ups which encompassed a tongue fixed to the turbine casing at this point. Thus the relative vertical movement between the centre of the casing and its ends altered the air gaps in the gauge assembly.

Figs. 16 and 17 show the apparatus and circuit diagram, and Fig. 18 shows typical records of hog and sag. The recorded movements in this case were small, the maxima recorded for all conditions being in the neighbourhood of 10–15/1000ths in. either side of the cold datum line.

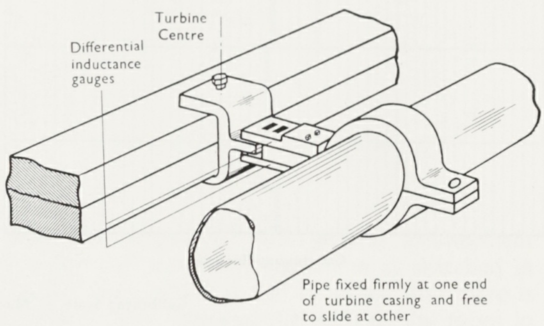


FIG. 16 Sketch showing attachment of gauges on H.P. turbine

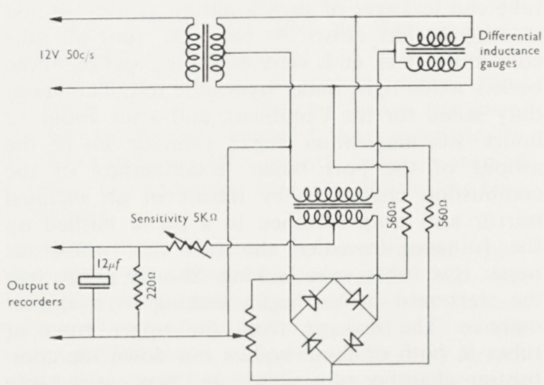


FIG. 17 Schematic arrangement of one recording channel, measuring the hog and sag of the H.P. turbine

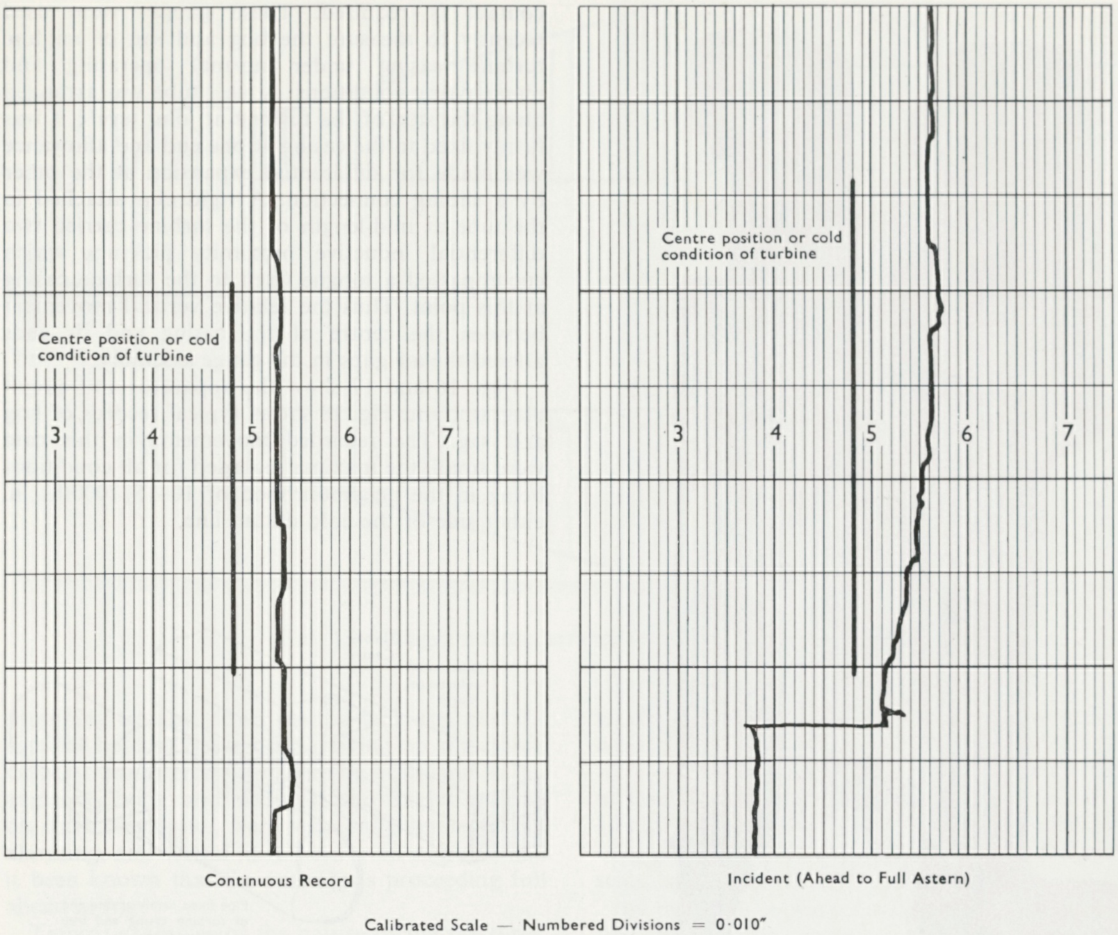


FIG. 18 Typical record of the hog and sag of the H.P. turbine

BOILERS

Back-End Leakages in a Scotch Boiler

A Scotch boiler investigation which left a strong impression on the Author's mind was one concerning severe back-end leakages. The vessel in question, with three boilers, was coal burning and had been dogged for some trips previously with tube end leakages of such a nature as necessitated shutting boilers down. At her U.K. port all tube ends in the back ends were expanded and the three boilers made tight under hydraulic test. The vessel duly sailed for the Continent, and after about 12 hours' steaming water began running out of the ashpits of the port boiler. Examination of the combustion chambers, by means of an inclined mirror and lamp attached to a plank pushed up the furnaces, revealed the fact that numerous upper row tubes were leaking. Shortly afterwards the starboard boiler began leaking in a similar manner. The leakage from the upper rows of tubes in both of these boilers ran down the combustion chamber tube plates, and was carried into the lower tubes; and in a very short time the tube nests, which were fitted with superheater elements, were choked to such an extent that the boilers would not steam and had to be shut down. The vessel's passage to the Continent was completed at very reduced speed on one boiler.

Internal examination of the boilers when they were sufficiently cool showed the tubes, plain and stay, to be fairly heavily encrusted with scale. Close examination of the tube ends where entering the combustion chambers showed this scale to be broken at the point where the tubes entered the back tube plate, as if they had been working in the tube holes and had retracted into the boiler on cooling.

Examination of the plans of these boilers showed that the end plates were in one thickness, with thick doublers riveted and welded to them in the steam space (*see* Fig. 19).

The main longitudinal steam space stays between these thick end plates (total $2\frac{1}{16}$ in.) were only 10 in. above the top rows of tubes and when, as happened in this case, the tubes became somewhat dirty on the water side, the difference in longitudinal expansion of the tubes, combustion chamber top and top row combustion chamber back stay assembly relative to the main longitudinal stays, put the top rows of boiler tubes in compression, resulting finally in the stay tube threads stripping and the top row tubes pushing their way through into the combustion chamber,

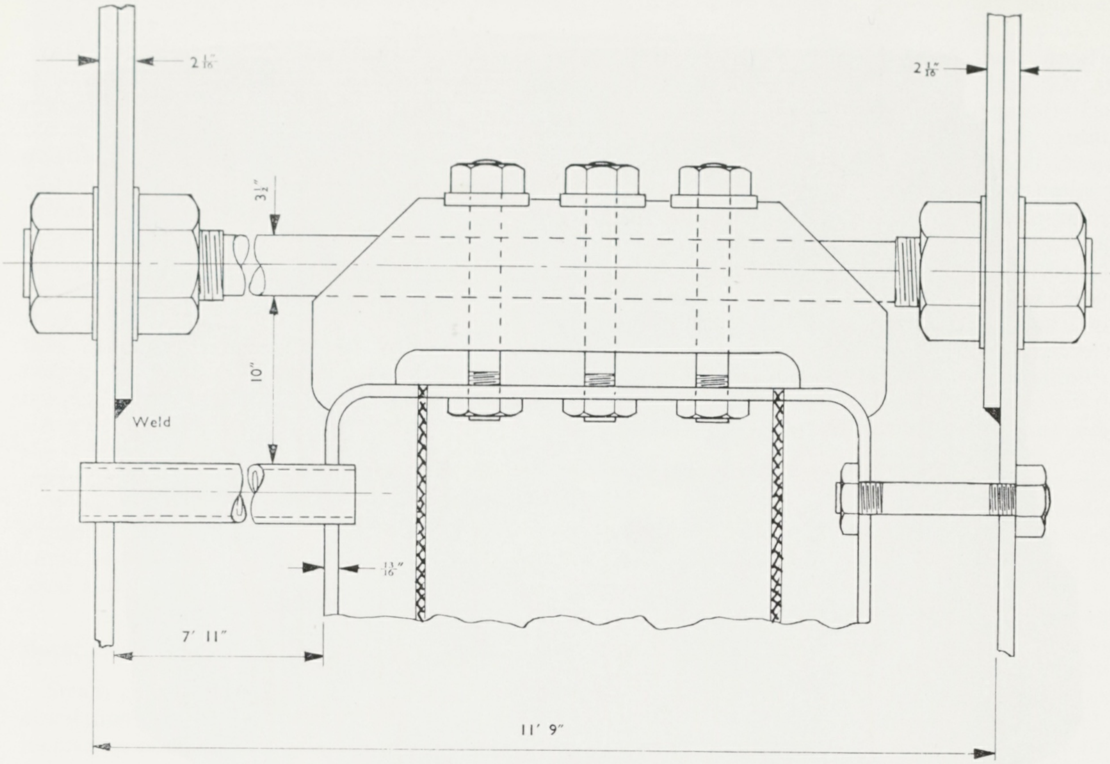


FIG. 19

causing the heavy leakages. That these boilers should have withstood a working pressure hydraulic test of tightness satisfactorily and, only hours afterwards, showed such severe leakages was little short of incredible. It was understandable, however, when one saw the condition of the upper tubes generally and realised the difference in expansion which must take place under steaming conditions, as opposed to uniform temperature conditions under hydraulic test.

Cleaning out all upper tube holes in the combustion chamber tube plates, including building up and re-cutting stay tube threads, and subsequently fitting new tubes completely cured the trouble, but it would no doubt have reappeared if these tubes, between such rigid supports, had again been allowed to become heavily scaled.

Failure of Waterwall Downcomer and Riser Tubes of "Victory" Class Babcock Boilers

This investigation was effected with a view to ascertaining, if possible, the cause of circumferential cracking through the full thickness of several downcomer and riser tubes in two war-time American-built Babcock & Wilcox header type boilers.

The cracks in question (*see* Fig. 20) were discovered when leaks in way of the expanded ends of these tubes in the headers were being investigated at a boiler survey. As will be seen from the photograph, these fractures extended for well over half the circumference of the tube end and were only just inside the expanded part—decidedly unpleasant shipmates! The general arrangement of tubes and waterwall headers is as sketched in Fig. 21, from which it will be noted that there is no access door opposite the ends of the tubes in question, which necessitates them being expanded with a special expander working at right angles to the tube axis.

Metallurgical examination revealed that the mechanical properties, structure and quality of the material were satisfactory. The cracks which had originated from the outside of the tubes were found to be transcrystalline, thus ruling out the possibility of caustic embrittlement.

Careful examination of the tube ends in question and others subsequently examined as a precautionary measure revealed the fact that the original expanding operations had been carelessly executed, inasmuch as in several cases, most noticeably that of the fractured tubes, the metal had been expanded excessively and eccentrically, the circumferential fractures all being in way of an excessively expanded area, eccentric to the bore of the tube.

In view of the foregoing it was considered possible that the expanding operation had left extremely severe longitudinal compressive stresses at the surfaces of the tube bores in way of the over-expanded areas, with correspondingly severe longitudinal tensile stresses on the outer surfaces. Such tensile stresses on the outer surfaces, if sufficiently high, could, it was thought, have initiated the fractures which had occurred, and it was decided to investigate this possibility by carrying

FIG. 20

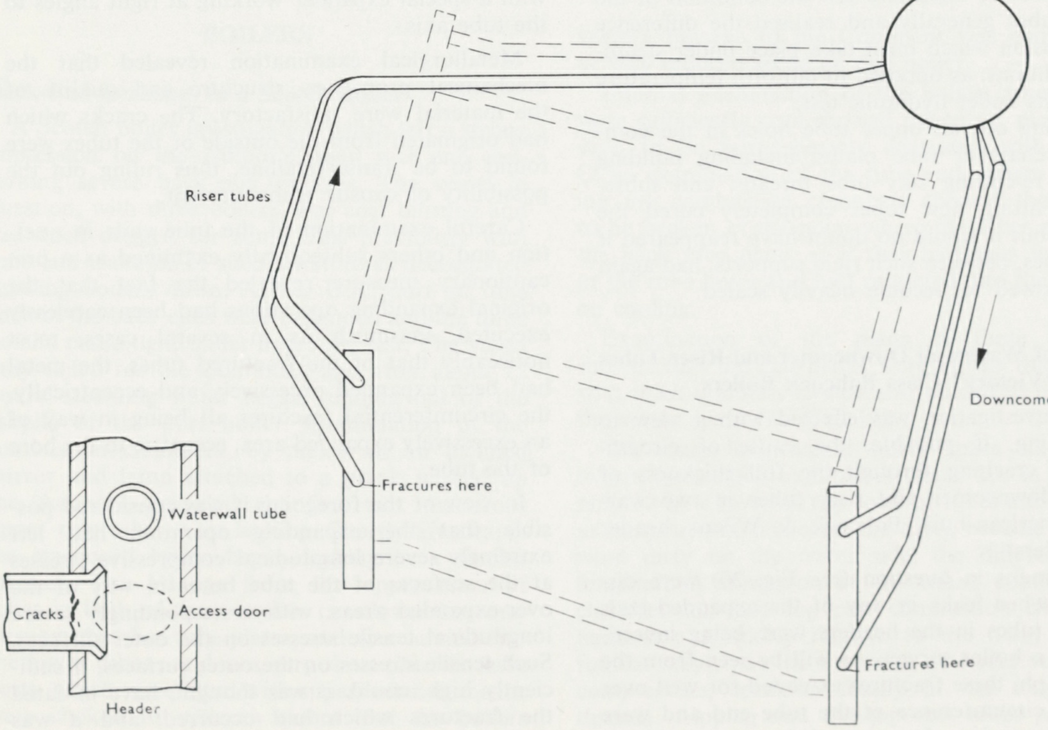

Riser tubes
Downcomer tubes
Fractures here
Fractures here
Waterwall tube
Cracks
Access door
Header

FIG. 21

out a strain gauge test under conditions simulating those of the original expanding operations.

For this test a piece of one of the original boiler tubes was expanded excessively and eccentrically into a tube hole in a piece of mild steel plate of the same thickness as the waterwall header. Subsequently this plate was cut away from the tube without damaging the surface of the latter, and resistance strain gauges were cemented to the outside of the tube in way of the expanded area. Zero readings were taken from the gauges, after which the tube was bored out in three stages; at each stage strain gauge readings were taken, the reduction in thickness being about 0·1 in. at each boring on a total thickness of 0·375 in. Final readings showed that a maximum longitudinal stress of about 20 tons tensile had existed towards the outer surface of the tube due to the expanding operations.

It was concluded therefore that through over-expanding, longitudinal surface tensile stresses could be induced of such a value as might initiate local surface circumferential failure.

Burnt Out Steam Generator

Steam generators or small self-contained forced circulation oil-fired coil boilers are being used in increasing numbers, particularly on board motor vessels where steam is continuously required for domestic or hotel services.

On one such vessel one of these generators (diagrammatically illustrated in Fig. 22) overheated whilst in operation, and apparently caught fire; this was with difficulty eventually quenched by hoses down the uptake. Subsequent examinations showed that the coils of the boiler had melted and literally burnt away, there being no visible escape of steam or explosion of any kind.

In an endeavour to ascertain why this failure had occurred, experiments were carried out, in co-operation with the makers, on a similar boiler at their works. It was demonstrated on several occasions that when the coil circulation was gradually restricted down to zero under full firing conditions, the steam pressure remained normal for a while and then the steam, with its temperature rising continuously, fell away in pressure to zero. Shortly afterwards the coils melted and burnt. Water applied down the uptake for quenching purposes was thought by some present at the time to stimulate the flames.

The one difference between the burning out of the boiler on board the vessel and the experiments carried out at the makers' works was the absence of possible soot accumulations in the uptake above the boiler, and believing that a soot fire was a necessary preliminary to any possibility of

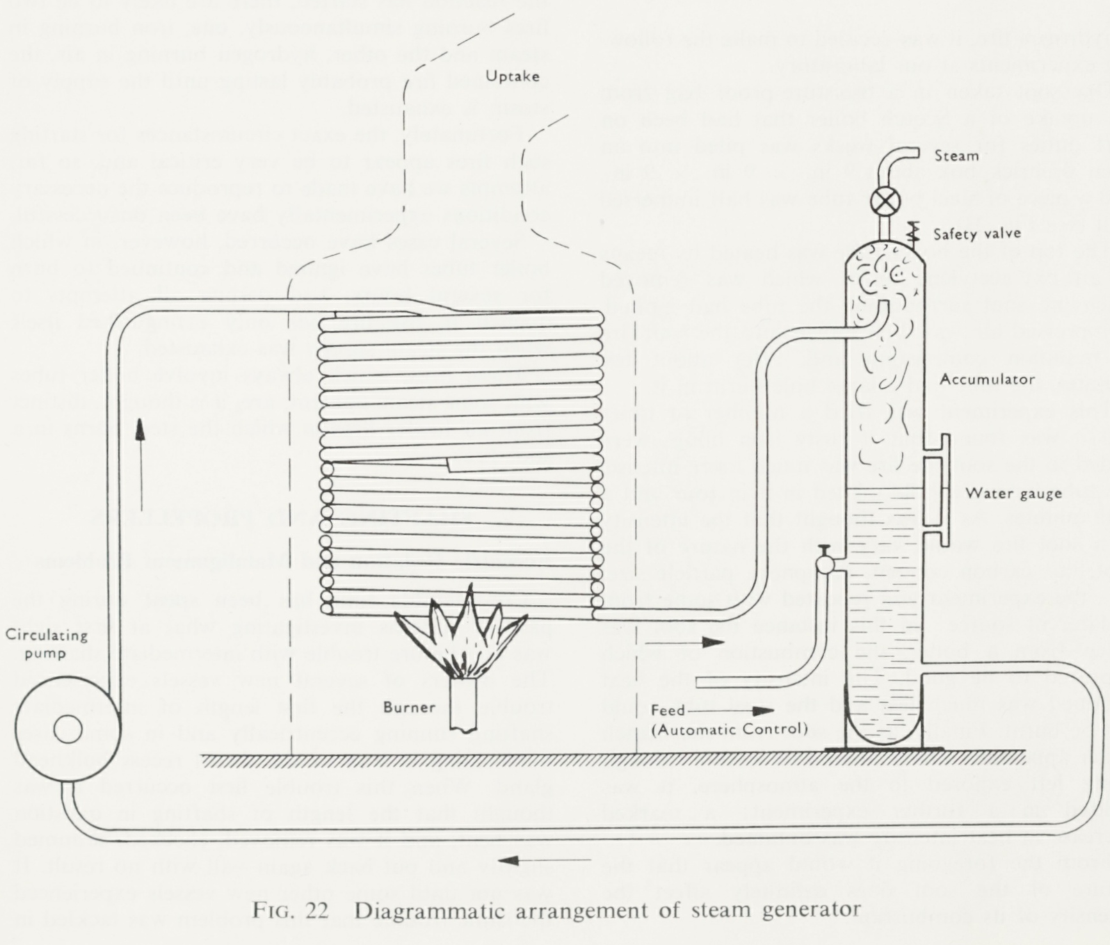


FIG. 22 Diagrammatic arrangement of steam generator

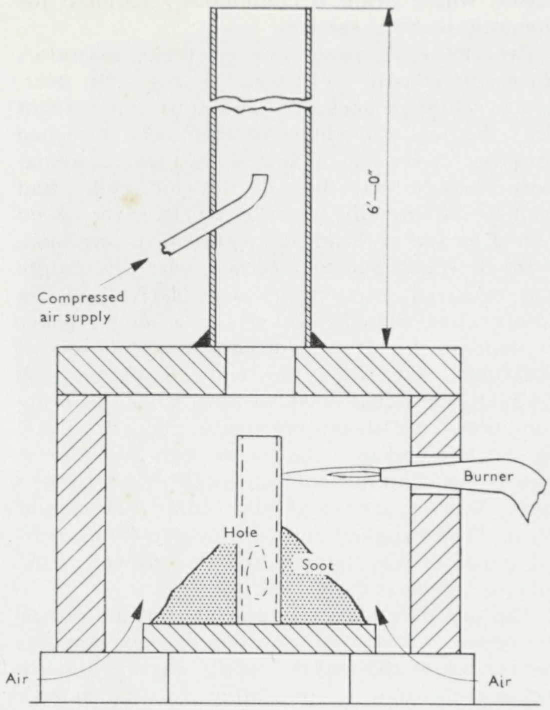


FIG. 23

a hydrogen fire, it was decided to make the following experiments at our laboratory.

Dry soot taken in a moisture-proof bag from the uptake of a Scotch boiler that had been on port duties for several weeks was piled into an open firebrick box about 9 in. × 9 in. × 9 in., and a piece of steel boiler tube was half immersed in it (*see* Fig. 23).

The top of the boiler tube was heated by means of an oxy-acetylene torch, which was removed when the soot surrounding the tube had ignited. Compressed air was then blown into the soot fire to maintain combustion and, after about ten minutes, the tube had a large hole burnt in it.

This experiment was tried a number of times and it was found that if rusty iron filings were added to the soot the fire was much more intense, the tube having a hole melted in it in four and a half minutes. As it was thought that the intensity of a soot fire would vary with the nature of the soot, i.e., carbon content, dampness, particle size, etc., the experiment was repeated with some from a different source: in this instance the soot was taken from a boiler, the combustion of which appeared to be good. The intensity of the heat obtained was much less and the steel tube could not be burnt. Finally, as the soot from the Scotch boiler appeared to have altered its texture through being left exposed to the atmosphere, it was utilised in a further experiment: a marked decrease in heat intensity was obtained.

From the foregoing it would appear that the nature of the soot does definitely affect the intensity of its combustion.

In the case of the steam generator previously mentioned, if the forced circulation pump were defective, the steam issuing from the coil would become superheated, the temperature increasing as the circulation decreased. Safety devices are now incorporated to prevent this occurring.

Similarly with some water tube boilers, when raising steam, it is possible if the superheater is not being sufficiently circulated to have very highly superheated steam in the superheater. If, under these conditions, due to poor combustion, deposits of soot rich in carbon have accumulated in the upper parts of the boiler, and are on fire, it is possible that oxides on the outside of the tubes take the place of the rusty iron filings, the forced draught takes the place of the compressed air and the tubes rapidly melt, as in the laboratory experiment previously described.

Dissociation of steam into hydrogen and oxygen in any quantity by *heat alone* requires temperatures in the region of 2,500° C. Iron will, however, burn in steam with the production of free hydrogen at much lower temperatures, ignition taking place at about 700° C. The iron will continue burning independently of any supply of oxygen from the air, and the hydrogen produced by the reaction will itself burn on coming into contact with air if the temperature is high enough to cause ignition.

This means that in the case of a boiler, once the reaction has started, there are likely to be two fires burning simultaneously, one, iron burning in steam and the other, hydrogen burning in air, the combined fire probably lasting until the supply of steam is exhausted.

Fortunately, the exact circumstances for starting such fires appear to be very critical and, so far, attempts we have made to reproduce the necessary conditions experimentally have been unsuccessful.

Several cases have occurred, however, in which boiler tubes have ignited and continued to burn for several hours, and despite all attempts to quench it, the fire has only extinguished itself when the steam supply was exhausted.

These fires, which always involve boiler tubes with some steam content, are, it is thought, distinct from air heater fires in which the steel burns in a soot fire.

SHAFTING AND PROPELLERS

Eccentric Rotation and Malalignment Problems

Considerable time has been spent during the past 12 months investigating what at first sight was an obscure trouble with intermediate shafting. The owners of several new vessels experienced trouble through the first length of intermediate shafting running eccentrically and in some cases overheating in way of the thrust recess bulkhead gland. When this trouble first occurred it was thought that the length of shafting in question was bent, and it was removed, checked, skimmed slightly and put back again—all with no result. It was not until some other new vessels experienced the same trouble that this problem was tackled in

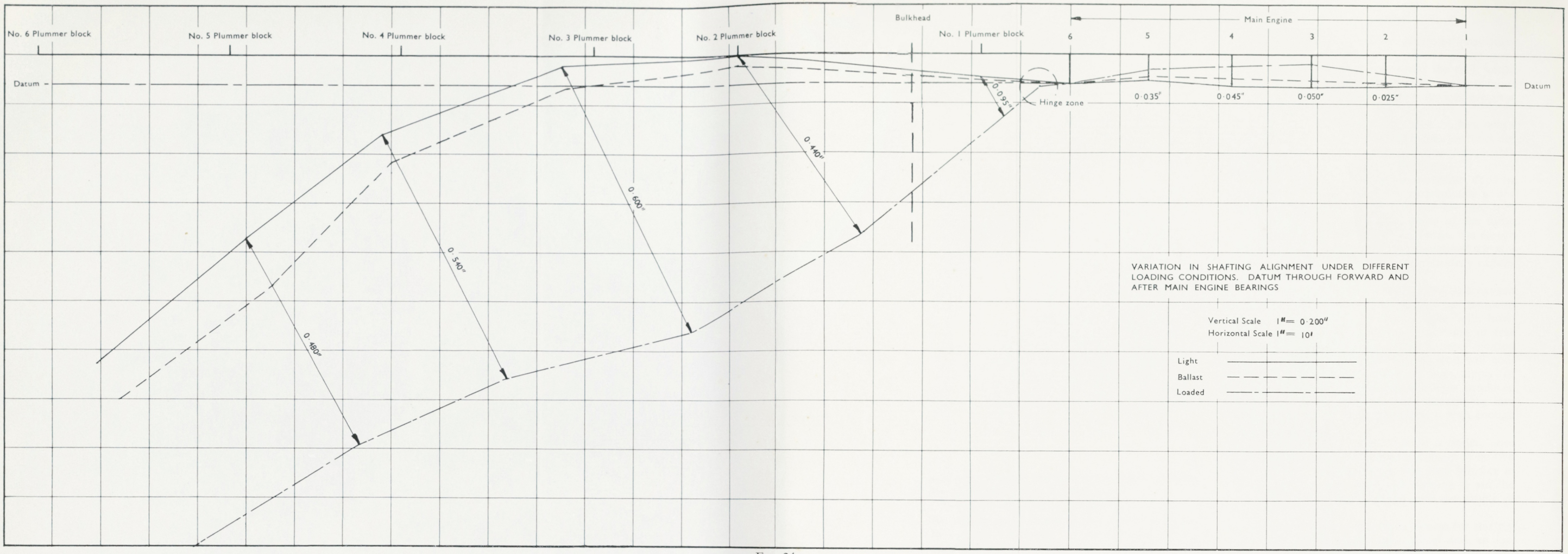

No. 6 Plummer block
No. 5 Plummer block
No. 4 Plummer block
No. 3 Plummer block
No. 2 Plummer block
Bulkhead
No. 1 Plummer block
Main Engine
6
5
4
3
2
1
Datum
Datum
Hinge zone
0·035"
0·045"
0·050"
0·025"
0·95"
0·440"
0·600"
0·540"
0·480"
VARIATION IN SHAFTING ALIGNMENT UNDER DIFFERENT LOADING CONDITIONS. DATUM THROUGH FORWARD AND AFTER MAIN ENGINE BEARINGS
Vertical Scale 1" = 0·200"
Horizontal Scale 1" = 10'
Light
Ballast
Loaded

FIG. 24

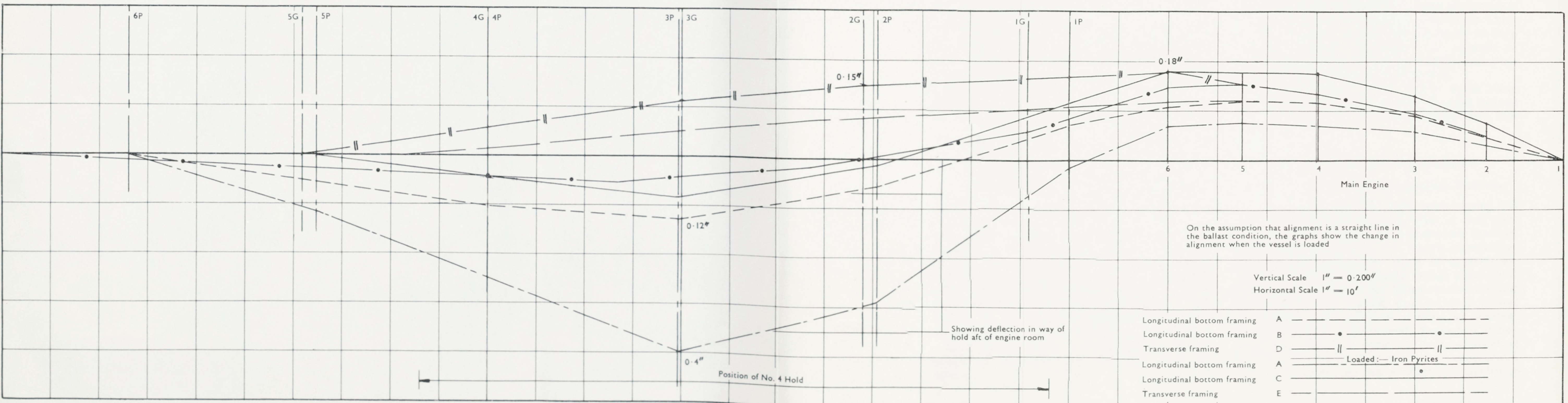

6P
5G
5P
4G
4P
3P
3G
2G
2P
1G
1P
0·18"
0·15"
0·12"
0·4"
6
5
4
3
2
1
Main Engine
On the assumption that alignment is a straight line in the ballast condition, the graphs show the change in alignment when the vessel is loaded
Vertical Scale 1" = 0·200"
Horizontal Scale 1" = 10'
Showing deflection in way of hold aft of engine room
Position of No. 4 Hold
Longitudinal bottom framing A
Longitudinal bottom framing B
Transverse framing D
Loaded :— Iron Pyrites
Longitudinal bottom framing A
Longitudinal bottom framing C
Transverse framing E

FIG. 26

earnest. The characteristics which could have any bearing on the problem and which were common to all the vessels were sorted out; likewise those which were common, but different from vessels of similar type, were noted. The characteristics common to all were longitudinal bottom framing, deep tanks aft of the engine room and 2SC opposed piston type machinery. Those common, but different from vessels of similar type, were in the ship's structure; namely, longitudinal bottom framing and amidship deep tanks.

Numerous shafting alignment checks were made on these vessels in loaded, light and ballasted conditions (a typical set is shown in Fig. 24), from which it was ascertained that the line of the forward lengths of the intermediate shafting to the crankshaft varied considerably under different conditions of loading, hinging about a point just aft of the main engines. That there was additionally, with difference in loading, a local varying deformation in the double bottom just aft of the main engines, was made obvious by the fact that under some conditions the No. 1 plummer block dropped clear of the shafting.

It was considered that as a first step to counteract the "throwing" of these intermediate shafts, measures should be introduced to ensure that the No. 1 plummer block never dropped clear of the line shafting, and this was effected by providing shims of suitable thickness for inserting when the vessel was in the loaded condition, with the proviso that they would be removed when the vessel was ballasted. This measure, although effective, was not exactly satisfactory, and ultimately local stiffening, having the effect of tying up the first plummer block to the engine seating, was fitted.

This investigation was interesting inasmuch as it would appear to indicate:—

(a) That there are still a lot of unknowns regarding hull bottom movements under different conditions of loading, with different types of framing, and with different ballast tank arrangements;

(b) that the mode of rotation of the 2SC opposed piston engine crankshaft journals, running in spherical bearings, can cause eccentric rotation of the intershafting if the first supporting bearing is too far distant from the crankshaft.

With reference to (a) above, it was decided, with the help of several shipowners, to endeavour to obtain a comparison between the support given to the shafting by longitudinal bottom framing and by normal transverse floors—typical cross-sections of each construction being as shown in Fig. 25 (a & b).

Investigations were carried out and for two vessels of each type, assuming the shafting to be in a straight line in all cases for the ballast condition, the effect of loading each is shown in Fig. 26.

The curves shown in this figure are impressive mainly on account of the scale used, and it is well to stop and consider whether, in line shafting, with which we normally have little trouble, $\frac{1}{8}$ in. in 10 ft. is of any consequence.

The most important aspect of the variation in alignment is that there should not be a large variation in that part of the shafting adjoining the crankshaft, either through different loading or in a seaway, as this imposes additional bending stresses on the after crank of the crankshaft. With this point in view it is always advisable to ensure that the main engine bedplate stiffness is carried on well aft of the machinery, at least to the first plummer block, and then tapered off gradually into the hull structure, to ensure a constant lead of the shafting to the main engine under all conditions.

At this juncture it should be mentioned that malalignment failures of aft cranks have occurred in aft end installations where, for torsional reasons, the diameter of the intermediate shafting has been considerably increased and the large barrel-like shafts have been supported in Michell pad type bearings. The crankshafts in question had in the course of some years worn down

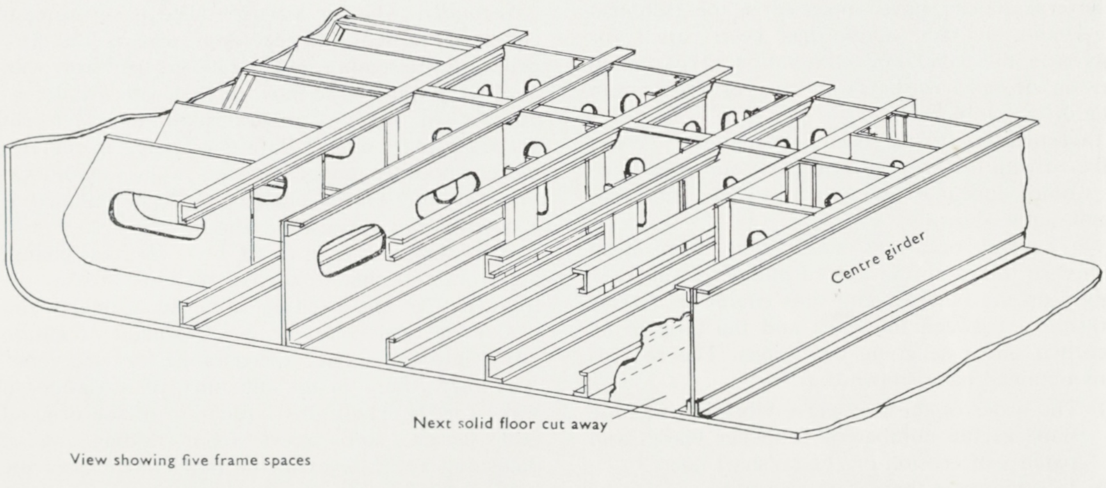


Fig. 25(a) Longitudinal bottom framing

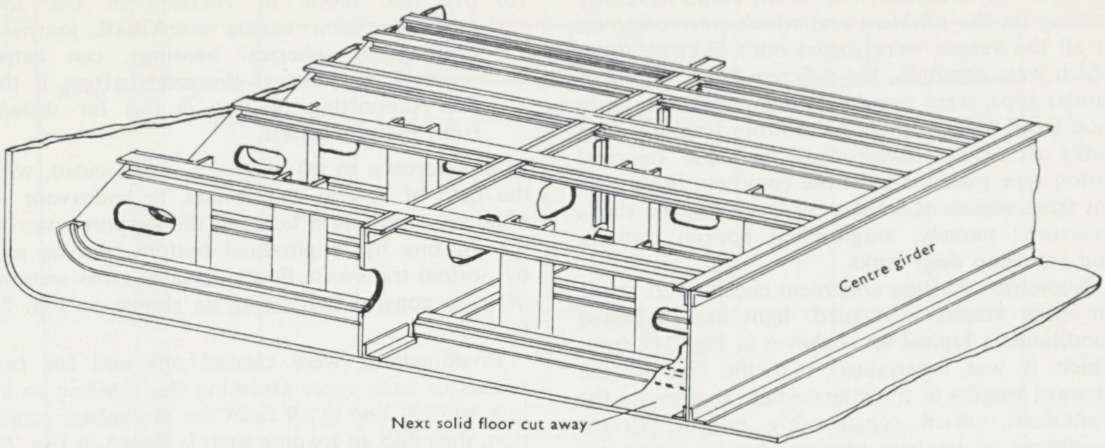


FIG. 25(b) Transverse framing

considerably, whereas the large, well supported intermediate shafts were still at their original level, with the result that the after crank deflections had become excessive.

Here again, I think it well to digress a moment, as when deflection readings are mentioned I feel it worthwhile to point out that in these days of stiff crankshafts, deflection readings are not such a valuable alignment indicator as they used to be. A case can be quoted where satisfactory crankweb deflection readings had been taken after fitting two new main bearing bottom halves; it was then requested that these bottom halves should be removed and the deflection readings taken again. This was done and the deflection readings with the bearings removed were found to be the same as those with the bearings in position. This means, of course, that if web deflection readings are to be relied on, it is imperative to ensure that the crankshaft is hard down in all its bottom half bearings when the readings are taken—this can sometimes be ascertained with special feelers but if there is any doubt, the shaft should be pulled or jacked down for each reading.

Eroded Tailshaft Liners

Several articles have appeared in the technical press and, in fact, action has been called for regarding the cause and elimination of patterned erosion of tailshaft liners in way of the lignum vitæ of the stern bush.

Investigations have been effected on vessels afflicted with this trouble, and from our experience it would appear to be practically confined to single-screw ships.

On several occasions stern tubes have been drilled so that a spring loaded plunger could bear lightly on the tailshaft and thus enable transverse movement between the shaft and the tube to be recorded under seagoing conditions. The records thus obtained have shown that: —

(a) The order of the transverse vibration was the same as the number of propeller blades and patches of erosion on the tailshaft liner.

(b) In some cases there appeared to be a resonant condition, whereas in others the transverse vibration was present all the way up the speed range, increasing in amplitude with revolutions.

It is not proposed to delve deeply into the cause of this type of vibration, except to say that in both (a) and (b) mentioned above the exciting force is the propeller, and that out-of-balance blade thrusts about the boss cause the propeller to be this exciting force. The out-of-balance thrusts are a function of wake speeds, and these in their turn are tied up with hull after-body design.

As in most investigations, once the cause has been found one has to specify remedial measures, and it is here, that in such cases one runs into difficulties.

Firstly, if as mentioned in (b), there is a resonant vibratory condition present near the service speed, this means that the propeller and tailshaft are vibrating transversely at their natural frequency about a point of support somewhere in the stern tube, and this point—and hence the natural frequency—will vary with weardown of the lignum vitæ, the variable quantity in this case being the weardown (this can of course be practically obviated by using a white-metalled stern bush and oil gland).

Secondly, whether or not there is a resonant condition present, the exciting force is the propeller. Re-designing the propeller, perhaps with the loss of some efficiency, so that with a different number of blades, greater skewback, different shaped leading and trailing edges, etc., it operates more smoothly under varying wake conditions, thus avoiding abrupt changes in thrust balance, is a possible but expensive remedy.

It must be borne in mind, however, that changing from four blades to five blades increases the frequency of the exciting force accordingly, and this in its turn could bring a resonant condition down into the service speed range, or vice versa.

Thirdly, there is the difficulty of variation in wake speeds. These are a function of the original hull design, and apart from adding water-deflecting appendages to the hull (which even the boldest would shrink from recommending), there is little that can be done once the vessel is in being.

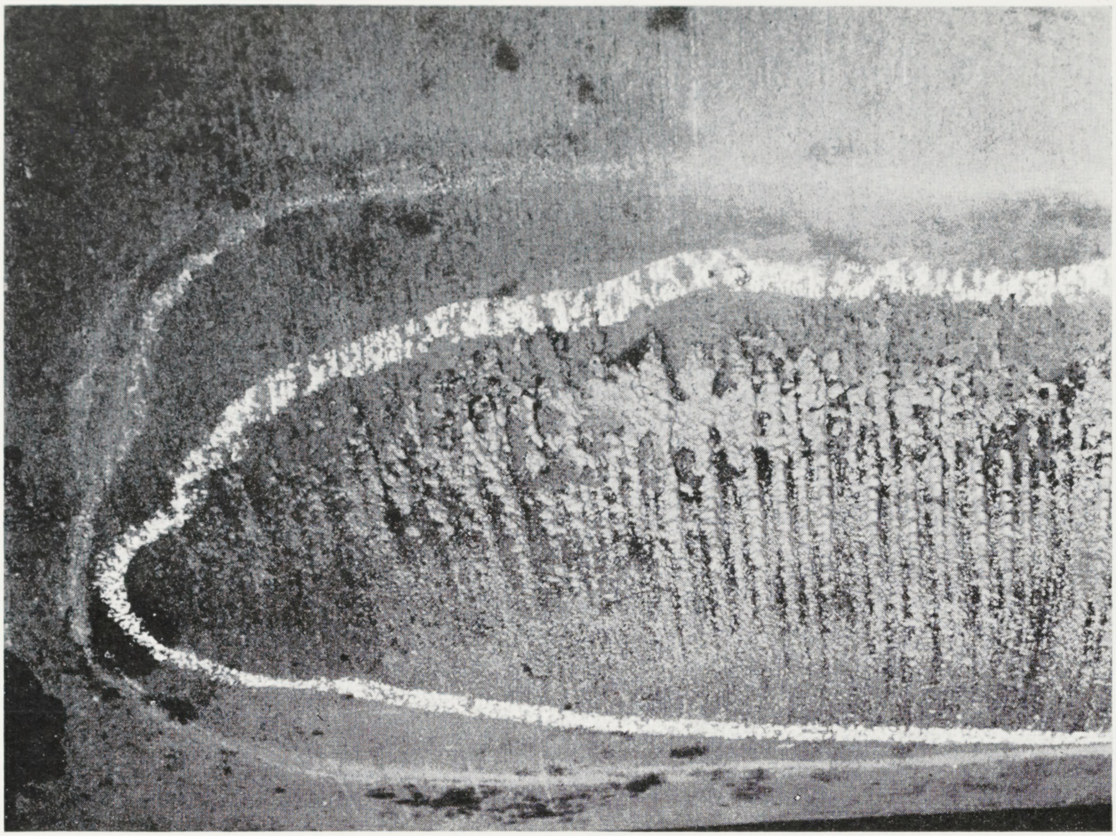


Fig. 27

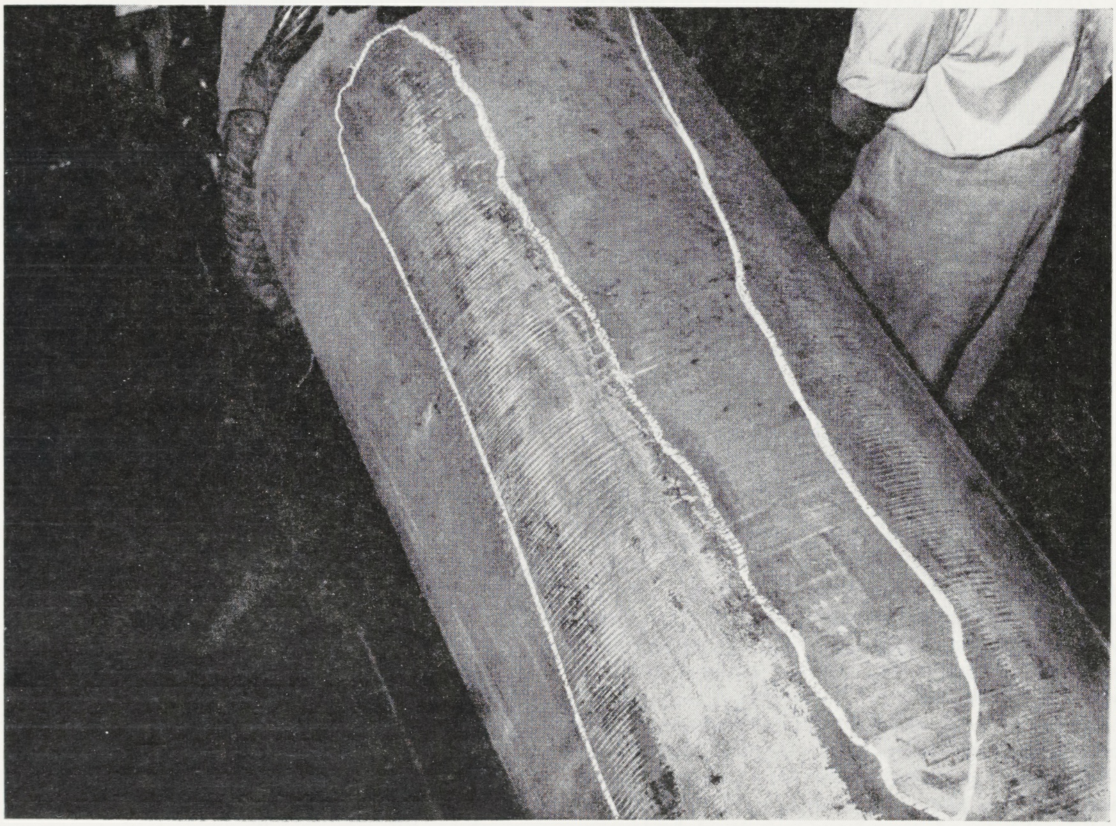


Fig. 28

Fig. 29

The foregoing is a very brief summary of some of the difficulties to be faced in solving this problem. The possible remedies mentioned are expensive, i.e., white-metalled bush with oil glands and/or re-designed propeller. If neither of these is possible, what can be done to extend the life of the tailshaft liner in such cases?

The erosion is as shown in Figs. 27 and 28, and it is thought that when transverse vibration of the tailshaft is taking place, the erosion occurs through:—

(a) Water being forced out under high pressure from between the wood and the liner.
(b) Water rushing into the vacuum formed when the shaft leaves the wood.
(c) A combination of the foregoing.

Obviously these effects will manifest themselves more fully towards the middle of the bearing surface area. When stern bush wood is first installed there are usually longitudinal grooves formed by the champhers on the abutting edges of the wood strips, but as weardown takes place these grooves are worn out, thus presenting an uninterrupted wooden bearing surface for the liner which is more conducive to the trapping of water or the forming of a vacuum when the shaft vibrates transversely.

It has been recommended in previous cases of patterned erosion that when re-wooding, circumferential water grooves (oblique to avoid ridges) about $\frac{1}{4}$ in. square and about one foot apart be cut in the wood to release any local hydraulic pressures or break down vacuum conditions which may be present through transverse vibrations of the tailshaft. The effectiveness of the measure is not known as yet on account of the length of time between tailshaft surveys.

It is well to consider what other defects, apart from patterned wear of the liner, can be produced by such transverse vibration of a tailshaft in its sterntube bearing. These are:—

(a) Rapid wear down.
(b) Excessive stern gland leakage.
(c) Trouble with the after tunnel bearing through hammering out of metal, and slackening or breaking of the bearing holding down bolts.
(d) Fracturing of the after end of the stern bush.
(e) Difficulties in keeping the propeller solid on the shaft.
(f) Fracture of shaft.

A further type of patterned wastage or erosion, again with a perfectly normal linered shaft and wooded sternbush, occurred in way of the neck bush at the forward end of the stern tube. These patches of erosion (*see* Fig. 29) were approximately in line with the propeller blades and in this case the action appeared to be galvanic.

The investigation was unfortunately not finalised before the vessel left for distant parts; the findings, however, suggested that:—

(a) There was a local galvanic action going on between the neck bush and the liner due to their different metallic composition—current

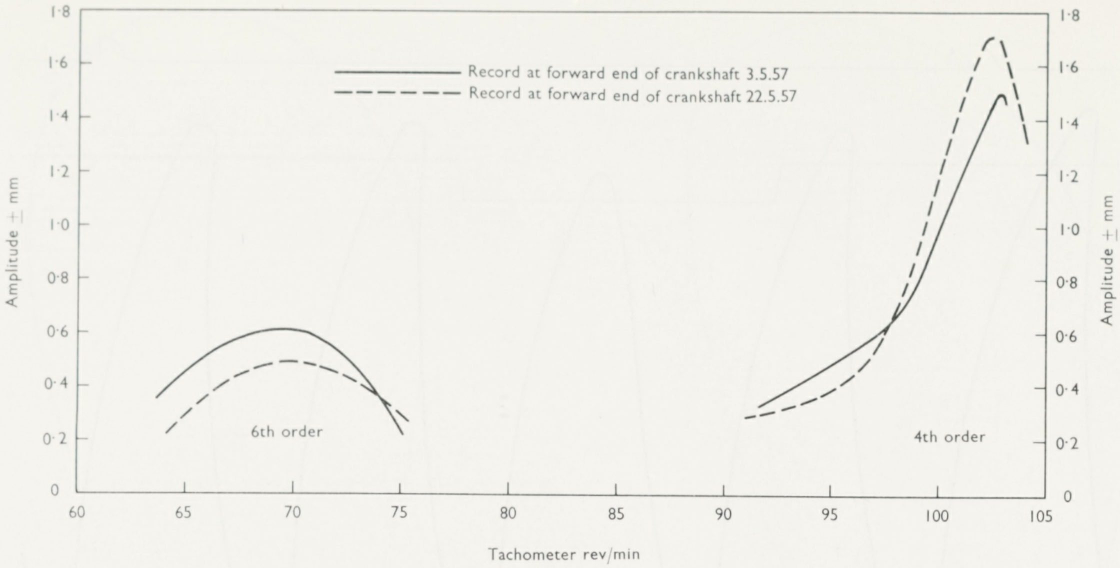


FIG. 30 Axial vibration

flowing and galvanic action taking place locally four times per revolution as the shaft, due to transverse vibration excited by the four-bladed propeller, rubbed the neck bush.

(b) The liner and neck bush rubbing four times per revolution completed the circuit between hull as anode and propeller with shaft as cathode, the current flowing then acting as an accelerator for the local galvanic action taking place in (a).

In this investigation potentials were measured between shaft and hull, and found to vary four times per revolution; the significance of this is not obvious as the potential could be varying with the blade proximity to the hull or the liner proximity to the sternbush.

Assuming the theories put forward in (a) and (b) have any foundation, it would appear that the remedy in such cases is to make the liner and neck bush of the same material, and after bonding propeller, tailshaft and hull together by efficient brush gear, to make the whole assembly cathodic by fitting efficient anodes on the hull.

VIBRATION

Axial Vibration of a Crankshaft

In the course of investigation work into the failure of a main engine crankshaft of a six-cylinder engine, axial vibration readings were taken at the forward end of the crankshaft. It was found that there was a sixth order vibration at approximately 70 r.p.m. of $\pm$ ·6 mm. amplitude, and a much larger fourth order at about 103 r.p.m. having an amplitude of $\pm$ 1·5 mm. (*see* Fig. 30). Although the propeller was four-bladed, there was a doubt that it was the origin of this fourth order. It was decided, therefore, to ascertain the effect on the axial vibration of altering the phasing of the propeller relative to the crankshaft.

Vibration records were taken on two coastwise passages of the vessel. During the vessel's stay in port between the two coastwise passages the propeller was turned ahead relative to the crankshaft by one bolt hole in a nine-hole coupling, i.e., 40°.

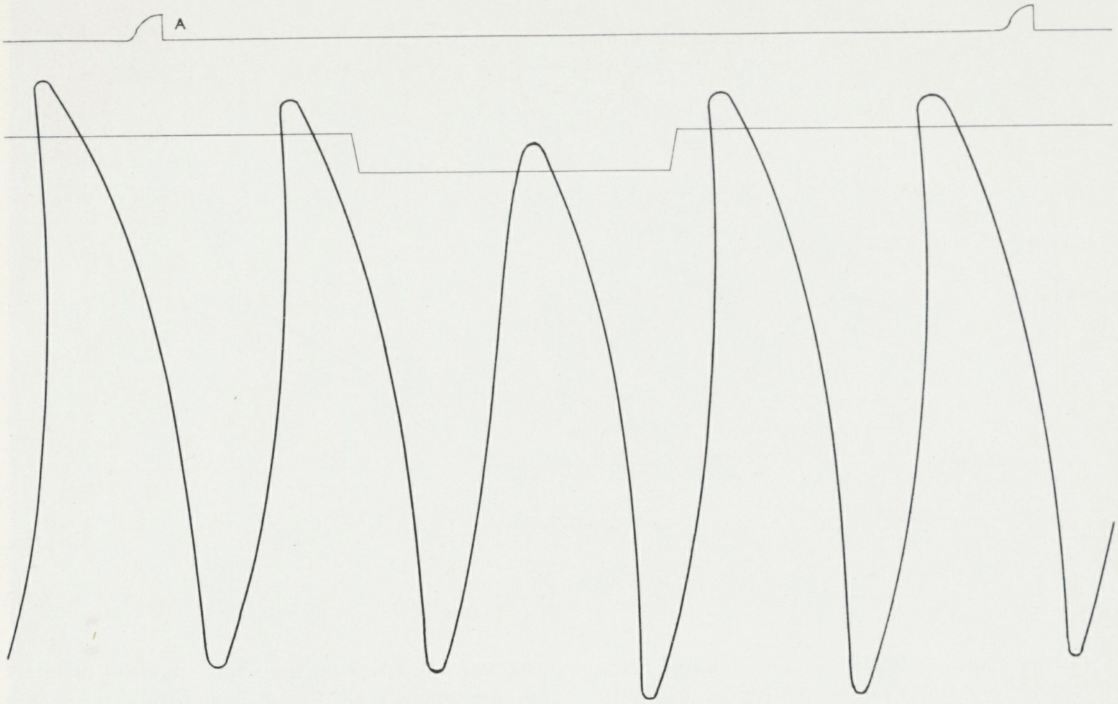
A sharp engine timing mark and a seconds timing mark were embodied on the records, and thus the r.p.m. and engine crank angles relative to the vibration impulses were known.

For comparison purposes two records were selected, one from each passage, at the same r.p.m., and from these it was seen that the phasing of the vibration impulses had changed relative to the crankshaft by an amount equivalent to the turning of the propeller, the amplitude being practically unaltered (*see* Fig. 31).

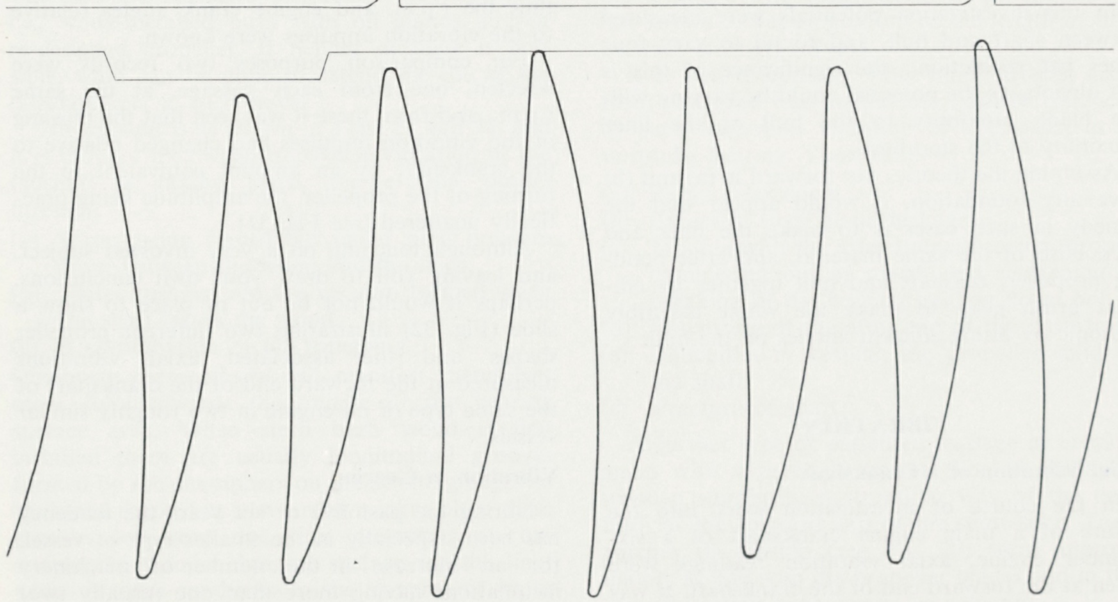
Although touching on a very involved subject, and leaving you to draw your own conclusions, perhaps it would not be out of place to show a slide (Fig. 32) illustrating two different propeller shapes and the associated axial vibrations measured at the forward end of the crankshaft of the same type of oil engine in two roughly similar vessels.

Vibration in Gearing

During the past five or six years the tendency has been, especially in the smaller type of vessel, for an increase in the number of machinery installations having more than one (usually two) main oil engines coupled by means of a gearbox to a common screwshaft. In most cases the gearing not only unites the outputs of the two prime movers but also provides a reduction in rotational speed so that it is possible to use moderately high-speed, compact units with consequent possible saving in weight and headroom.

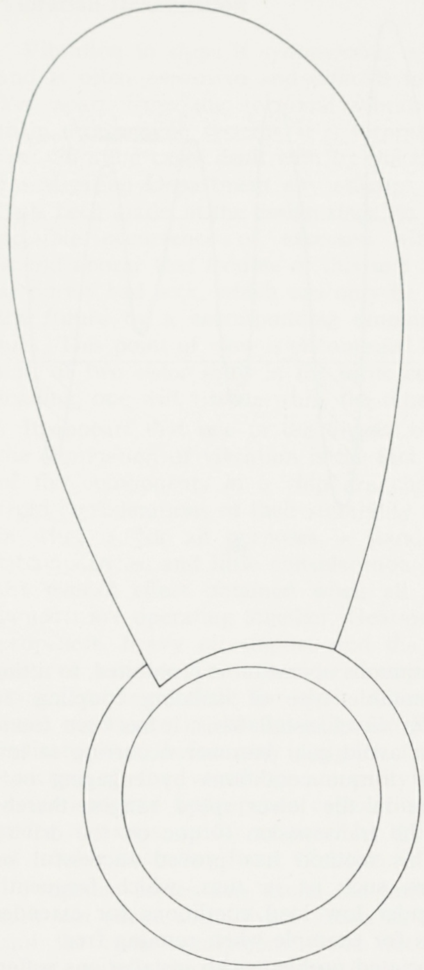


Record taken before shift.



Record taken after shift.

FIG. 31 Enlargements of Records (not to scale) showing shift of phase of impulses relative to engine timing mark A, when propeller turned 40° ahead relative to crankshaft, r.p.m. same in both cases



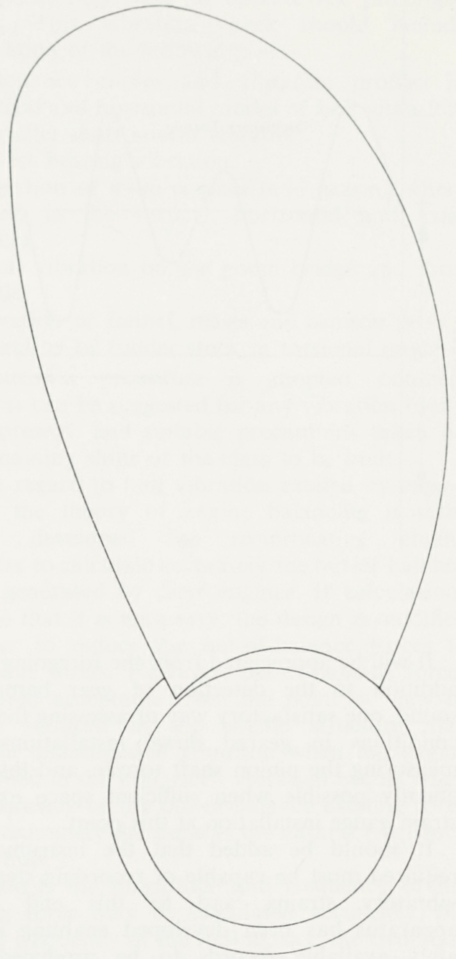


FIG. 32 Vibration amplitude 2·5 mm. approx. Vibration amplitude 1·5 mm. approx.

The general undesirability of operating gearing from a source of power, such as a diesel engine, where an oscillatory torque component may exist, is well known; the oscillating torque component may, especially at lower power conditions, exceed the transmission torque and result in cyclical reverse loading of the pinion and wheel teeth in the reduction gear. The resultant shock loading which occurs to the teeth is usually known as "gear hammer" as it is frequently accompanied by audible impacts between teeth. It has been usually accepted practice, therefore, to "isolate" the main engines from the gearing as far as oscillatory torques are concerned by fitting some form of coupling; this reduces the torsional natural frequencies of the system, or provides additional torsional damping, or—and this is the more usual case—fulfils both of these requirements.

Typical couplings are hydraulic and magnetic slip couplings—the latter do not introduce much additional damping but provide extremely high flexibility. More recently other types of hydraulic coupling have been developed which in general provide lower slip and consequently higher efficiency, and which introduce a controllable degree of damping by adjustment of the flow of operating fluid. These couplings are not as effective in removing torsional critical speeds from the operating range as other types due to their lower flexibility. They will reduce severe resonant torque fluctuations at critical speeds but sometimes transmit non-resonant torque fluctuations, with the result that in certain circumstances gear hammer is present at low operating speeds where the transmission torque is low.

To investigate the torsional vibration conditions of such an installation it is necessary to employ more elaborate instrumentation than with a direct drive system—a mere measurement of torsional swing as is given by a torsiograph is not usually sufficient, as the torque conditions in the pinion shaft must be known both as regards steady and oscillatory components. Fig. 33 shows graphically oscillatory torque such that (a) load reversal will not occur, (b) load reversal will occur. Tooth separation will occur when at any instant the net torque changes from positive to negative or vice versa.

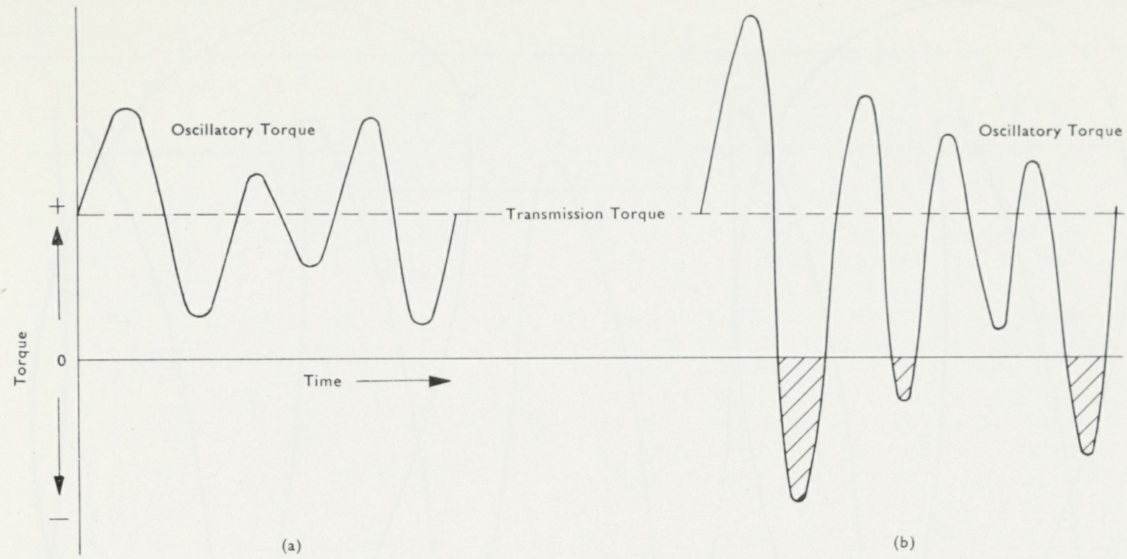


FIG. 33

It will be appreciated from the foregoing that, in addition to the detection of gear hammer by sound, one satisfactory way of assessing the torque conditions in geared diesel installations is by measuring the pinion shaft torque, and this is frequently possible when sufficient space exists for strain gauge installation at this point.

It should be added that the instrumentation required must be capable of recording steady and vibratory strains, and to this end suitable apparatus has been developed enabling immediately available records to be produced under operating conditions. A reproduction of a typical record taken on a single engine geared installation is shown in Fig. 34.

Measures taken to improve the operating conditions in geared diesel installations vary from merely stipulating a barred speed range over which continuous operation is prohibited, to fitting a more suitable type of isolating coupling. In certain twin diesel installations, it has been found possible to avoid gear hammer occurring at low speed, low torque conditions by engaging only one engine in the lower speed ranges, thereby doubling the transmission torque on the driving pinion. This method has proved successful on installations such as in tugs, which frequently operate under low load conditions for extended periods, as for example when running free.

An associated problem with installations suffering from gear hammer is that when the gears are designed to operate with relatively large backlash, a spread of the torsional critical range has been observed in some cases, resulting in the necessity for a wider restriction of speed range than might otherwise have been necessary.

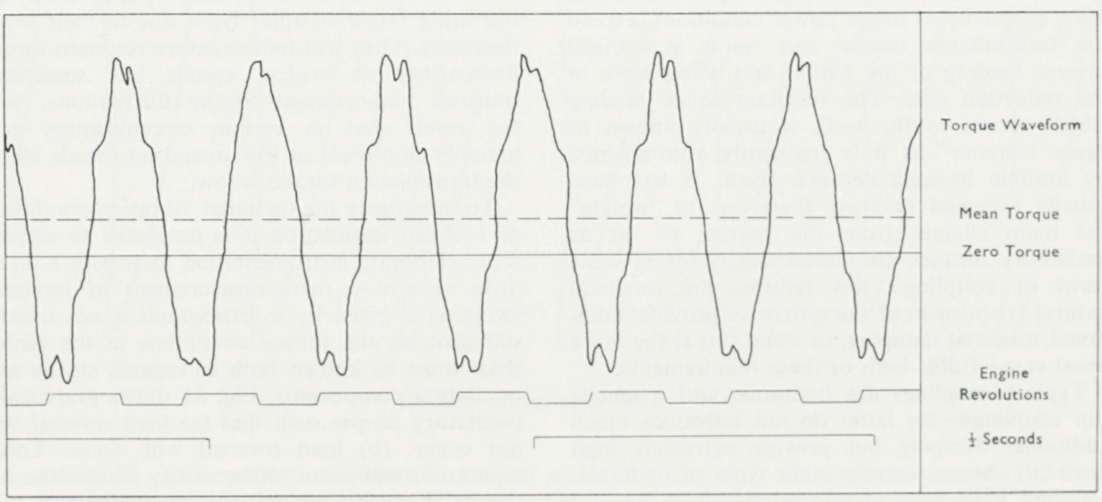


FIG. 34 Strain gauge record from pinion shaft

Vibration Investigation

Vibration in ships is synonymous with trouble and is often expensive and difficult to eradicate. Yet apart from the torsional vibration of the main transmission systems, it is surprising in how few vibration cases dealt with by the Engineering Investigation Department any attempt appears to have been made, at the design stage, to predict the possible occurrence of excessive vibration. It would appear that trouble of this sort is regarded as purely bad luck, which can only be avoided in the future by a corresponding amount of good luck. This point of view is encouraged by the fact that of two sister ships in the same condition of loading, one will vibrate while the other will not.

It appears that one of the biggest obstacles to the elimination of vibration is the fact that many of the components in a ship are chosen solely from considerations of their suitability to perform in what is, for all purposes, a standard set of circumstances, and little consideration is given to the overall effect obtained when all such components are operating together. Gearing, turbines, propellers, heavy oil engines and the hull itself are, of course, first considered separately by specialists responsible for their selection. However, experience in our Investigation Department indicates that, before any design is finalised, the concept of the vessel as an entity should be scrutinised and appropriate calculations made in order to assess, as far as possible, the likelihood of trouble occurring from vibration.

Propeller forces can excite the hull, which in turn can excite such diverse items as the gearing and the radar equipment. It is not surprising, therefore, that during vibration investigations one is told by the propeller makers that they have many similar propellers operating in similar vessels without trouble, or by the gearing manufacturers that similar gearing is operating quite satisfactorily in such and such a vessel, and one is asked by the radar designer "What else but a breakdown can be expected when the set is shaken from morning till night?"

The various components may do all that is claimed for them, they may be well made and entirely in accordance with their specifications, but if, for instance, the natural frequency of the radar set in its mountings can be excited by main engine forces, it is very likely that trouble will occur.

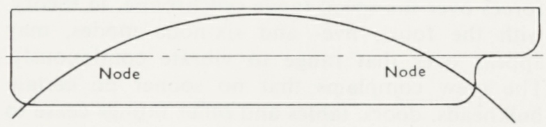
It is not desired to give the impression that it is possible to calculate accurately the natural frequency of a gearcase on its seating, or a bridge structure, but attempts have been made and data is being slowly accumulated on the subject. This also applies to forces exciting vibration. Calculations must be compared with and supported by observations and measurements on actual vessels, especially when several vessels of one class are to be constructed. In this case, not only should an estimate be made of the frequency of the fundamental modes, but the first vessel of the class should be checked for vibration in the loaded and light ballast conditions, and any other condition in which it may have to operate for prolonged periods. This vibration check should include examination of the following:—

1. Resonance curves and vibration profiles in vertical and horizontal modes of hull vibration.
2. Propeller shaft lateral vibration.
3. Thrust bearing vibration.
4. Vibration of main engines (and gearing where fitted) in the vertical, horizontal and axial modes.
5. Local vibration on the poop, bridge and forecastle.
6. Vibration of funnel, masts and samson posts.
7. Vibration of rudder stock in torsional mode.

If such a procedure is adopted, countermeasures can be suggested for any vibration found to be present, and suitable precautions taken on the remaining ships of the class to be built.

With regard to hull vibration excited by engine forces, the theory of engine balancing is sufficiently developed for reciprocating engine designers to calculate accurately the out-of-balance forces generated by their engines. If calculations indicate that it is necessary, the design is modified in order to reduce the out-of-balance forces to acceptable limits. However, out-of-balance forces considered acceptable from the point of view of stresses induced in the engine structure may generate unacceptable vibration when the engine is installed in the hull. Such vibration occurs when the frequency of the out-of-balance exciting force is approximately the same as the natural frequency of the hull.

Although the magnitude and frequency of the exciting force may be calculated accurately, the natural frequency of the hull in, say, the two-node vertical and two-node horizontal modes (Fig. 35) varies with the loading of the ship.



Two-node vertical

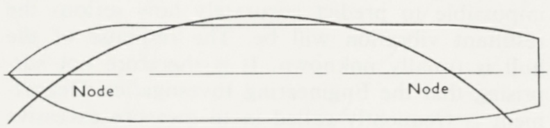


FIG. 35 Two-node horizontal

It is, however, possible to estimate these frequencies for any given condition of loading, and this should be done for all new tonnage. If it then appears that the natural frequency of the hull and exciting forces might coincide in, say, the fully loaded condition at full power, the force itself may be reduced by means of balance weights or, as these criticals are usually quite sharply tuned, the

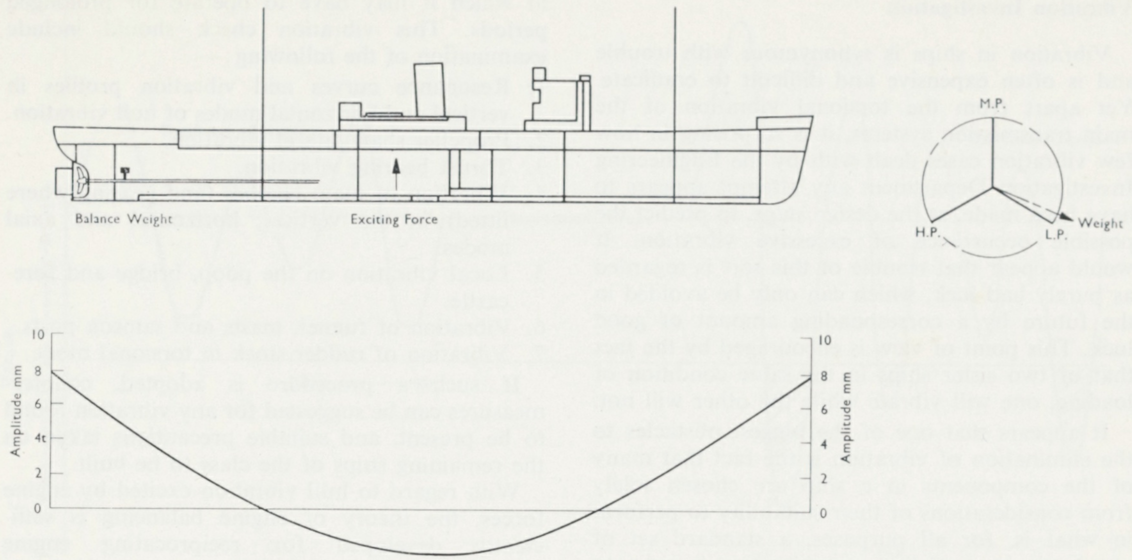


FIG. 36 Profile of two-node vertical hull vibration

propeller pitch may be modified so that the vessel will operate at an engine speed clear of the criticals. It might also be possible to reduce the effect of the force by altering the fore and after positions of the engine, although generally such a measure would prove prohibitively expensive except in the earliest design stages.

The calculation of the natural frequencies of the hull for three- and four-node and higher modes of vibration is less certain; the interval between each natural frequency becomes less sharply defined the greater the number of nodes, so that in practice, a vessel in which exciting forces over the speed range synchronise, in theory, with the four-, five- and six-node modes, may appear over that range to vibrate continuously. The crew complains that no sooner do certain bulkheads, doors, tables and other fittings cease to vibrate than others commence.

But even if it is possible to calculate the probability of coincidence of the natural frequency of the lower modes of hull vibration and the frequency of some exciting force, it is virtually impossible to predict accurately how serious the resultant vibration will be. The response of the hull is usually unknown. It is therefore not surprising that the Engineering Investigation Department is frequently called in to measure excessive hull vibration generated by "small" out-of-balance engine forces, and recommend measures for its avoidance or elimination.

As previously stated the response of the hull to a small exciting force is more or less unknown, so much depending on the damping capacity of the hull and any cargo carried, and the fore and aft positions of such a force. A small exciting force near an amidships antinode can set a whole vessel vibrating; on the other hand, when such conditions exist, a superimposed opposing force of a relatively smaller magnitude acting at the after antinode where the amplitude is generally much greater, can be used to "kill" the vibrating system.

This was demonstrated very dramatically at a recent investigation—the principle involved is not new and the remedy, although effective, does not at first sight appear to be sound engineering. This case concerned a cargo vessel 420 ft. long which vibrated excessively in the two-node vertical mode at a frequency which varied from 78 c.p.m. in the loaded condition, to 87 c.p.m. in the light ballast condition, both being at much used engine r.p.m.

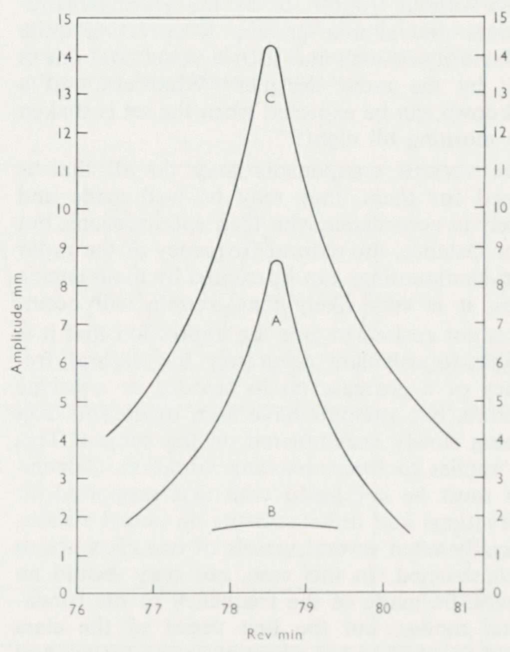


FIG. 37 Resonance curve from poop

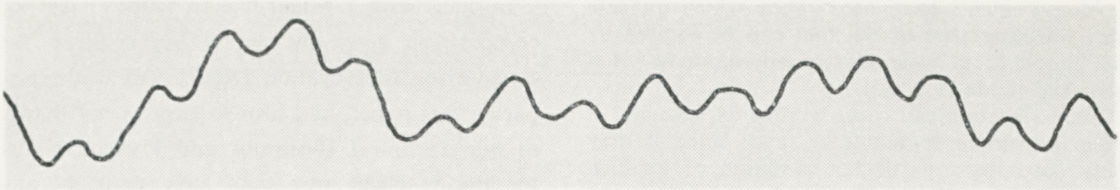


FIG. 38 Vibrograph from poop

Calculations revealed a primary out-of-balance vertical force in the engine of approximately seven tons; since the engine was located at an antinode, it was concluded that this force was exciting the vibration. Fig. 36 shows the position of the engine in the ship and the shape of the vibration profile.

It was decided to fit a counterweight on the intermediate shafting near to the stern gland. Owing to limitations of space, it was not possible to use a weight as large as simple theory indicated, but a weight of 780 lb. with an effective radius of 19·4 in. was used. This generated a force of 1·18 tons at 78·6 r.p.m.—the location and angular position of the weight are shown in Fig. 36.

Fig. 37 shows vibration resonance curves taken on the poop. Curve A gives the amplitude generated by the out-of-balance force in the engine when no counterweight was fitted to the intermediate shaft. Curve B gives the amplitude when the force generated by the counterweight is in direct opposition to the exciting force. The vibration is reduced to negligible proportions. After the readings for curve B were obtained, the counterweight was rotated 180° on the shaft so that it assisted the exciting force. Curve C indicates that the ensuing amplitude was more than 14 mm., i.e., a total movement of 28 mm. or just over 1 in. on the poop.

Most superintendent engineers will no doubt shudder at the thought of a balance weight whirling round on their tunnel shafting, thinking of the extreme wear and tear on tunnel bearings, its effect on alignment, not to mention possible hazard to personnel. The fact remains that a relatively small balance weight, strategically placed, will in such cases reduce a vibrating liability to a smooth running machine.

Fig. 38 shows the original vibration as recorded on the Geiger vibrograph. The small, evenly spaced fluctuations are due to the vibration and the larger, irregular movement was caused by the slight pitching of the vessel.

The example given above is relatively simple, and a simple and inexpensive solution was quickly forthcoming, but as the size, speed and power of vessels increase vibration problems become more complex. There are occasions when part of a ship, whilst possessing sufficient strength, has a stiffness value which causes its mass to vibrate against the rest of the hull mass. The part may be relatively unimportant, like a bulkhead, or it may consist of the entire bridge structure (Fig. 39).

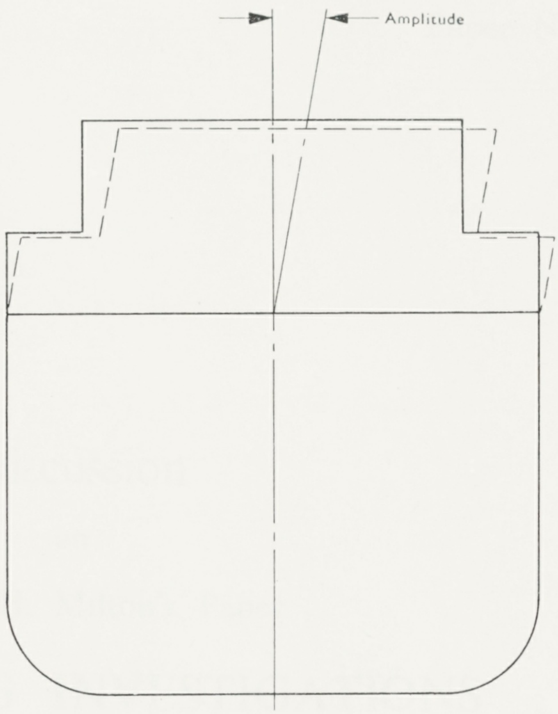


FIG. 39

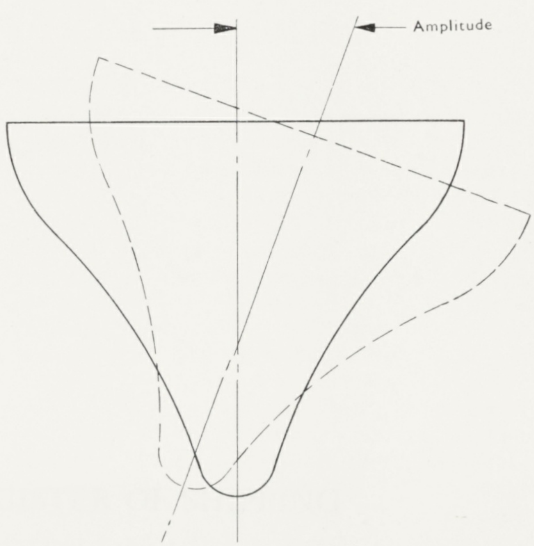


FIG. 40 Stern vibration torsional mode

Measurements have been taken which indicate that the after part of the hull can be excited by the propeller in what closely approximates to a torsional mode (Fig. 40).

Such vibration can cause serious damage and is often expensive to eliminate, and, since it may only occur in a particular condition of loading, may not be encountered until a vessel is many months or even years old.

In conclusion I would like to acknowledge the co-operation given by certain Surveyors in the Engineering Investigation Department whilst preparing this paper, and also to express my thanks to our Technical Illustrator and Printing House for the excellent way they have prepared and presented the diagrams and text.

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Discussion

on

Mr. J. H. Milton's Paper

INTERESTING INVESTIGATIONS

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

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Discussion on
Mr. J. H. Milton's Paper

Interesting Investigations

Mr. S. ARCHER

Those of us who have known and worked with Mr. Milton for any length of time will, I feel sure, agree that in this admirable paper he has once again demonstrated his mastery of the difficult art of "Special Investigations". That to succeed demands art as well as engineering science may be disputed by some, but reading through this fascinating account of a wide range of marine engineering problems and their solution, one is impressed as much by the Author's skill as an investigator, as by the power of the methods adopted.

The examples presented are very well chosen, as much for the insight which they permit into the methods of approach adopted, as for the actual solutions achieved. If I may say so, Mr. Milton has an uncanny knack of worrying a problem from every likely angle, eliminating all the various possibilities one by one, until he has reduced it to its simplest terms, and in many cases the solutions are astoundingly simple and practical. The case of the bottom end bearing oil box in Fig. 9 is a good example.

I would therefore commend his advice in the fourth paragraph of page 1, where he urges that possible normal causes of a given trouble be investigated *before* embarking on more complicated or theoretical approaches.

If I were asked what four qualities would be most valuable for engineering investigation work, I would choose the following:—

1. Keen observation.
2. A questioning mind.
3. The ability and determination to measure, and
4. Power to interpret such measurements in terms of a satisfactory solution.

All these qualities are, in my opinion, clearly evident in Mr. Milton's paper.

He has also throughout pointed to the need for fundamental and/or applied research on a number of subjects where present knowledge is deficient. It would be interesting to know, in say ten years' time, how much of this much-needed research will have been completed.

Altogether, the Association is fortunate to have this valuable contribution to its Transactions.

Mr. Milton's advice that the simple possibilities should first be explored reminds me of a puzzling case some years ago. The triple-expansion engines of a tanker suddenly began to suffer from top and bottom end bearing trouble in the L.P. "leg" with recurrent overheating and wiping. The Society was called in to deal with the problem and, as there seemed to be no obvious cause, careful investigations were made, firstly, on the bearing loading, including the calculation of polar diagrams, also bearing design with special reference to the number and disposition of oilways, etc. At the first opportunity, shaft alignment checks were made but without result. Next, vibration tests at sea on bedplate and seating yielded equally negative conclusions. During these tests, however, it was noted that the after L.P. journal was throwing in its bearing a matter of 10–15/1000 in., but it was not until later that its significance was realised. Had we observed keenly enough, we ought to have spotted that the L.P. forward crankweb was actually bearing on the after face of the forward L.P. main bearing shell. It was finally discovered that during a previous routine overhaul the main thrust ahead pads had been interchanged with the astern set and the difference in axial thickness had been sufficient to allow the crankshaft to move forward as described. The L.P. crank was thus being "bowed" under the full propeller thrust, consequently "throwing" the rod with unfortunate results. Simple enough, yes, but . . . !

The case had its somewhat humorous aspect, however, since, when no clear solution was forthcoming, mutual recrimination started between the hull and engine departments of the shipyard, culminating in an accusation by the latter that the ship must be "too weak locally in way of the L.P. engine"!! Reluctantly, after much argument and despite an opposite opinion from the Society, the yard actually fitted extra fore-and-aft stiffening in the form of side girders port and starboard in the double bottom in way of the after end of the engine. The cost must have been quite considerable! History does not relate what the shipbuilders said to their engineering colleagues after it was all over, but it should have been worth listening to!

In Fig. 3 showing records of the vertical deflections of the "A" frames in a Doxford engine taken at sea, it would appear that the period of the long waves was about seven seconds. This was typical of pitching in a seaway and could the Author confirm that, in fact, these periodic strains arose from that cause? If that were so, it reinforced a long-held suspicion that Doxford bedplates, seemingly requiring lower fore-and-aft bending stiffness than engines not working on the opposed piston principle, tended to follow hull movements more easily in a seaway. This might account, at least in part, for the considerable incidence of fractured guide plate bolts in certain Doxford engines. These guide plates constitute a fair proportion of the total fore-and-aft strength of the engine structure and are, of course, arranged at the rear of the engine between "A" frames, the axes of the securing bolts being fore-and-aft. It is significant that most of the failures

reported have been of the direct tension type. Would the Author agree that this could be a possible explanation?

The Author's practical hints on bearing design and adjustment were valuable and doubtless carried an explanation for many a "mysterious" hot bearing and broken bottom end bolt. It was agreed that wherever possible an attempt should be made to measure the diametral distortions consequent upon hardening-up the bolts, but, of course, this would be difficult with main bearings unless the crankshaft were first removed.

In the example of the twin screw, four-crank, triple-expansion, steam-reciprocating machinery with the defective seatings, worn chocks, etc. (Figs. 11 and 12), could the Author state whether the engines were fully balanced (e.g. Yarrow-Schlick-Tweedy), or only partially balanced? Experience had shown that one of the most common causes of slack, fretted and worn holding-down chocks, etc., was the presence of a large rotating unbalanced couple, often introduced deliberately to reduce resultant primary vertical unbalance and thus minimise resonant excitation of a possible 2-node vertical mode of hull vibration. The likelihood of exciting 2-node horizontal hull vibration was much less, this natural frequency in most ships being some 50 per cent or more higher than for the 2-node vertical.

The case of the back-end tube leakages in the Scotch boiler (Fig. 19) was probably not uncommon, especially where heavy scaling of the top rows of tubes was allowed to occur. The Author seemed to suggest that the vertical spacing between underside of lower main stay and the top of the tube nest, viz.: 10 in., was abnormally tight. However, examining the plans of some half-dozen boilers of similar size and type at random, it was found that in fact this dimension varied between about 9 in. and 11 in. I would therefore suggest that the main cause was simply the heavy scaling of the tubes, although possibly the stiffening effect of the rather heavy top doubling of the front and back end plates could have contributed to the trouble.

The investigation of the burnt-out steam generator (Figs. 22 and 23) was very revealing. Would it be right to conclude that the self-supporting form of iron combustion, independent of air, calls for the prior supply of steam or moisture as a prerequisite? In any case it seems clear that the "nigger in the woodpile" is the presence on the tubes of sufficient amounts of soot of a particular composition, probably associated with iron oxides, to supply the necessary high temperatures required to initiate burning of the tube wall.

The particular examples of shaft (or should one rather say "hull"?) misalignment and eccentric rotation of shafting (Figs. 24, 25 and 26) were disturbing, in that one normally assumed longitudinal framing would give greater insurance against such troubles than transverse framing. Could it be that when loaded with a dense cargo, such as iron pyrites, concentrated near the centre line, this could have set up transverse bending moments in the double bottom which the transversely framed ships would be better able to resist? The other important lesson to be learnt from these troubles, as pointed out by the Author, is clearly the importance of ensuring a reasonably constant lead of the shafting to the main engine crankshaft under all loading conditions.

In certain 4-cylinder Doxford engines, especially those in aft end installations, there had been a definite pattern of crankshaft failures involving usually the forward side web of No. 4 crank. Did the Author think these failures could have been influenced in any way by the "trundling" effect of the shafting to which he had referred (in the verbal presentation)?

On the subject of shaft couplings between heavy oil engines and gearing (page 25) it should be pointed out that one of the types whose limitations are described by the Author, has recently been extensively redesigned with automatic increase of slip (pneumatically controlled from the engine fuel lever) as the input speed falls below a predetermined value. This has the effect of preserving a reasonably low, and thus economic, slip at normal service speed whilst providing increased damping from the higher slip in the lower speed range where mean transmission loads on the main reduction gear teeth are small, and which would otherwise permit severe gear hammer at major torsional criticals resonating at these speeds. Recent sea trials by the Society's E.I.D. on a large Dutch-built ocean-going tug have successfully demonstrated the effectiveness of the redeveloped coupling, using electrical resistance strain gauges on the pinion shafts.

Among the interesting examples of hull vibration described by the Author my attention was caught by the torsional mode of stern vibration (Fig. 40). Torsional oscillation of ship's structures is probably not very common, but I do recall one case of a small naval vessel during the war, on which I measured resonant torsional amplitudes on the forecastle head 180 degrees out of phase between port and starboard wings.

Mr. J. B. Davies

Mr. Milton has given us a most interesting paper and I am very sorry that I will not be able to attend the meeting as I will be away in Germany.

In his introduction Mr. Milton makes one statement which summarises a difficulty we frequently run into. He says "investigations are effected and information obtained, but because no previous research has been done—no just comparison can be made". Unless we know what stress or deflection we get in a successful ship, or component, it is not possible to know whether the measured stress or deflection in the unsuccessful ship is the cause of the trouble or whether we must carry the investigation further to find the real cause.

It is, I suppose, natural that Owners are generally willing to allow investigations on ships with

troubles, but much less willing to provide comparable facilities on trouble-free ships. A comprehensive series of investigations on such ships would be most valuable.

The last two paragraphs of the remarks on "Main Engine Seating Trouble" remind me of a vibration investigation made some 14 or 15 years ago. The main engine and everything around it was certainly vibrating, but it was some time before anyone looked at the holding-down bolts and found a lot slack.

There can be no doubt that the Author is correct when he says that there are still a lot of unknowns regarding hull bottom movements and I am still not at all certain as to just why we had shafting alignment problems in the particular ships mentioned at the bottom of page 18. The measures taken seem to have cured the troubles and I have not heard of any other ships running into this particular problem. As the Author says, is $\frac{1}{8}$ in. in 10 ft. of any consequence in a ship which is itself constantly deflecting as a wave passes along its length? I sometimes envy the bridge builder who excavates down to solid rock before starting his foundations!

In the section dealing with vibration investigations the Author remarks that few builders attempt to look at the ship as a whole from the vibration aspect until trouble is experienced. He may be interested to know that several builders are now asking us to carry out such an investigation before the design is finalised. I might add, however, that they are mostly builders who have had serious troubles, and it would be as well if other builders would do the same and not wait for their own troublesome case to occur.

Mr. G. D. DUNKEL

Fig. 18 of this paper shows a recorder trace of turbine casing deflections. The "incident" depicted by this recording suggests a rapid transition from sag to hog of the casing, the time scale being dependent on paper speed. Both recorders mentioned in this paper have variable paper speeds of which the slowest is $\frac{1}{2}$ cm./sec. Mr. Jennings has informed me that this record was taken on a Bailey strip chart recorder with a chart speed of approximately two longitudinal divisions per hour. However, even at this slow speed, one must conclude that the chart was stopped for this incident.

Mr. J. GUTHRIE

Mr. Milton has presented a most readable and enjoyable paper on investigations to marine machinery, and the Staff Association could do with a few more like it.

As a layman in the field of investigation, I must tread warily in discussing some of the cases reviewed in the paper, but there are several points not very clear to me. In the research into the Doxford engine bedplate deflections, the Author describes the use of a steel strip 2 in. wide by $\frac{1}{8}$ in. thick threaded through the lightening holes and attached to each end of the bedplate. Now, in my youth, I had a little jingle drummed into me from a 19th century mathematician, to the effect that:—

"And so no force, however great,
Can stretch a cord, however fine,
Into a horizontal line
That shall be absolutely straight."

This steel strip, according to the rhyme, would sag into a catenary, and being supported between two points on a vibrating bedplate, this catenary should, in theory, vibrate according to its natural period independently of the bedplate. How then, can the relative movements of these two vibrating objects—the bedplate and the steel strip, as recorded in Fig. 3—be considered purely as bedplate deflections?

There is another case connected with bedplates: that of twin-screw steam reciprocating machinery where the results of the investigation required the removal of both engines for seating repairs, realignment and re-chocking. The Author states that it was obvious that the engines themselves were moving on the tank top, and later, after an examination of the double-bottom tank, that holding-down bolts were loose, intercostals were cracked, and top angles were slack. As this sort of trouble must have been going on for years to be as bad as it sounds, one can only hope the vessel was not classed. Incidentally, what was wrong with the Superintendent, or the ship's Chief Engineer, when they were unable to see what was so obviously wrong with their engines?

Mr. Milton's notes on burnt-out steam generators are most interesting, and his conclusion that soot accumulation is the probable cause of the fires may be correct. My own experience of hydrogen fires in boilers is confined to two cases: both occurred shortly after the war, both were on extra high-pressure boilers and in both instances the vessels had been lying up for an undetermined period. The first was on Hitler's yacht-destroyer *Güille* which was being steamed from a North European port for delivery to a Near-Eastern potentate. After a hydrogen fire in one of the Benson boilers, causing a total breakdown of the machinery, she was towed to some port in Italy (I believe) for conversion. Here, the damage to the boiler was complete, there being precious little left of it to explain the cause of the fire, which had extended to various auxiliaries in the boiler room.

The second case was on a German-built cargo vessel which had spent several years undergoing survey for L.R. classification in Mexico, and which developed hydrogen fires in both Wagner boilers only a few days after completion of survey, when about half-way across the Atlantic. Upon examination, both boilers appeared intact, the air casings and steampipes being complete and unscorched. The boiler drums appeared in order, but, in the case of the starboard boiler, the tubes were found to be charred to the extent that they could be snapped between the fingers, for all the world like sticks of charcoal. The port boiler had

suffered a similar fire, but not so extensive, yet both boilers had separate uptakes and there appeared to be no means whereby the fire from one boiler could have been communicated to the other.

One of the most interesting cases in the paper concerns that illustrated in Fig. 36 where a counterweight is fitted to the intermediate shaft near the stern gland. This is indeed an unorthodox remedy, and one can imagine the amount of persuasion required to convince the average hard-headed Superintendent that such a weight would reduce vibration. One can also imagine the watch-keeping engineers' language when they had to tighten the stern gland!

Finally, this reminds me of a question I have longed to put to E.I.D. personnel: when an Owner is put to considerable trouble and expense in re-chocking and/or re-aligning the machinery to cure vibration, what happens if there is no improvement?

Mr. B. HILDREW

As the paper ranged over a very wide field, to discuss the varied problems fully would take a very long time and accordingly it is proposed to limit comments to one or two items. The Author's introductory remarks on lack of research in certain engineering designs are noteworthy. In this context a recent investigation showed that not even the manufacturers of soft stern-gland packing know the rubbing speeds and operating conditions which limit the life of the article sold by them.

The further comment that the simple solution should always be looked for is also particularly relevant. In offering such a solution, one often hesitates to charge a fee, despite the knowledge of the heavy cost incurred by the customer in trying to solve the problem, particularly as some customers consider simple solutions ought to be given free. The elegant simplicity of the re-designed oil box outlined in the paper as a solution to overheated bearings is a classic example and such a solution because of its simplicity can give great personal satisfaction to the solver.

The section of the paper on vibration investigation is a summary of the variety of problems occurring in this field. The Society now offers a service to Shipowners and Shipbuilders, to investigate the possibility of hull vibration at the design stage. To date, experience shows that such a service can only be advisory. As an example, four sister ships proposed for a continental Owner have a secondary unbalanced couple of 350 tons/ft. The two- and three-node hull frequencies can generally be determined reasonably accurately and in the hull form proposed it is known that in the 3-node mode of hull vibration which will occur at 2 $\times$ full speed engine r.p.m. the engine will be on an anti-node. The primary forces and couples and secondary forces are balanced.

A reduction of secondary unbalance can be effected, but the consequent introduction of primary unbalance in the engine would excite the 2-node vertical mode of hull vibration at full speed. Alternative methods of secondary balance using synchronous motors or gearing are expensive and often unacceptably noisy. The Society has had to advise the Owner of the possibility of hull vibration but has suggested that he should accept this risk and install the engine as designed.

Another unusual case arose some years ago when an apparently balanced Burmeister and Wain engine was fitted amidships in a dry cargo ship. The vertical internal couples of the engine were found to excite the hull in the 2-node mode. An easy solution to this problem proved to be non-existent. The size of weight on the line shafting can only be determined by experiment and in the context of the ship in question proved unacceptably large, preventing access to the stern-gland. Since this ship the Shipbuilder has worked out all vertical, horizontal and torsional hull modes for his new construction and related them to any engine unbalance in the machinery installation. A recent calculation carried out by him at the design stage has shown the same problem might occur again and an approach has been made to the Society to suggest a solution. The Society can only advise that the engine seating should be stiff enough so as not to deflect under the loading imposed by the internal couples as in such circumstances the ship should not excite. Accordingly, consideration is being given to the adequacy of the engine seating in this context. Of course an alternative solution would be to examine the possibility of installing another engine design.

These examples show that even at the design stage a simple solution to a ship vibration problem is not always possible and certainly support the Author's contention that such early investigations are desirable.

Mr. A. R. HINSON

Mr. Hinson thought that the Author had covered a diverse collection of ship and engine investigations in a manner which was indeed suitable to the title of the paper. He had made them interesting.

Most of the examples, although apparently dissimilar, had all called for fundamentally the same technique. The investigation of bedplate deflections in a seaway, Fig. 1, and dynamic stresses in a crankshaft, Fig. 5, and practically all vibration problems usually proceed in three easily recognisable stages.

Instead of discussing the effect of inertia on stretched tape and such-like details, it might be of more value to discuss these stages generally.

In stage one, a defect occurs. This would appear to be a necessary prerequisite for an investigation, but experience has shown that this is not so. Defects can be imagined, their symptoms can appear and then disappear; and—and here's the rub—they can be reported owing to confusion of the unusual with the undesirable.

The first stage then, is to ascertain that a defect actually exists. There is no problem if a component has fractured, or a bearing has wiped. But

if the defect is not so obvious, the investigation can lose momentum and degenerate into a more or less interminable controversy. The writer once flew to Gibraltar with a Superintendent to board a ship which was allegedly suffering from a knock in the engine. Speed had been reduced to five knots. Once clear of the Straits—a condition exacted by the Master—speed was increased to full power, whereupon the engine ran like a sewing machine for three days. The Superintendent was sarcastic. He tapped the desk with his forefinger and asked the Chief if the knock sounded like that. The writer, having had a good look round without finding anything unusual, retired to the wireless room to compose a cable. It did not read well. While the writer was thinking it over the deck shook.

The knock in the engine had been described so vividly there was no mistaking it. It banged again when the writer reached the engine room floorplates. Speed was reduced. The knock continued and at each bump everything shook. Crankcase doors were removed and the job was felt by hand like a steam reciprocator. The knock seemed to move from one end of the engine to the other. Then it ceased, and to the chief's obvious satisfaction could not be made to start again. It was not until the engine was partly dismantled that certain strained studs indicated the source of the noise. If a decision had been taken during the first three days of the investigation, it probably would have been a wrong one.

In stage one, the defect is studied to establish that it is not a particular symptom of a general condition; a symptom which, if merely suppressed, will either reappear elsewhere, or worse, deprive the Shipowner of a valuable alarm signal. It has been known for a hot bulkhead gland to result in the hull being stiffened.

In stage two, the problem is defined. This is probably the most important part of an investigation. It is here that the vibrographs, electronic pick-ups, strain gauges and alignment gear can be used to profit. And if it is true that at this stage the specialist, working to prove or disprove the engineer's or naval architect's hunch, is particularly valuable, it is also true that without the hunch the equipment is useless.

Before installing elaborate instrumentation, it is prudent to ask oneself if it will be possible to base any decision on the results. The answer may be "no", for whatever is being investigated may be so badly damaged that the results are invalidated.

The Society's plans approval, metallurgical and other departments often assist in defining problems. In the case illustrated in Fig. 4c in the paper, the torsional vibration department were able to show that such a fracture could only have resulted from overspeeding. It was subsequently found that this had, in fact, occurred.

Stage three, effecting a cure, is usually the easiest stage of all. If the problem has been defined correctly, the recommendation is not so much the result of profound reflection as the statement of an obvious fact. This is the reason for the curious sense of anti-climax which often accompanies the completion of a difficult investigation. In the beginning, when all is confusion, and red herrings litter the trail, the case holds a challenge. Unfortunately, as in other things, familiarity breeds, not contempt, but bathos. Perhaps this is inevitable. If so it is a pity, for it affects the morale. The only satisfactory antidote is to start work on another problem before the current one is quite finished.

Mr. J. M. Jennings

I enjoyed reading Mr. Milton's paper very much, bringing back as it did memories of being fogbound in Flushing and stormbound in the Irish Sea!

I would like to make a small correction to the details about the Dessyn gauges used to measure engine bedplate deflections. The instruments as supplied by the manufacturers are not potentiometers. They were adapted by us for this purpose in order to operate them into a graphic recorder. The operating range of the instruments was 0—0·1 in. and not 0—0·5 in. Although I agree with Mr. Milton that the records obtained from motor ship bedplates were perfectly satisfactory and no errors arose from tape vibration, it is difficult to convince others that this is so, and moreover the fitting of the tape and Dessyns is difficult and the whole thing heavy to transport. We are developing an ultra-sensitive low frequency accelerometer which would be more satisfactory. The accelerometer consists of a strain-gauged cantilever with a mass on the free end. An acceleration normal to the plane of the beam will produce a strain at the root of the beam proportional to that acceleration. The resultant strain gauge signal can be electrically integrated so that it will then be proportional to the displacement of the instrument. The sensitivity is achieved by using a new type of strain gauge, the strain sensitivity of which is 60 times that of a wire resistance strain gauge. The new gauge consists of a filament of semi-conductor germanium or silicon similar to the materials used in the construction of transistors. This material, which has a high inherent strain coefficient, can be made either to increase or decrease its resistance with a positive strain applied. The high sensitivity means that the accelerometer has been designed so that the beam has a natural frequency well above the frequencies encountered in ship work, and responds down to 100 c.p.m. without requiring amplification or an even lower frequency with amplification. The accelerometer is so sensitive that we have yet to find anyone capable of holding it in the hand steady enough to produce no measurable trace on our recorder.

The semi-conductor gauges can be used for measuring thrusts in bearing blocks and tension in tug towing gear, where a large safety factor has been applied to the design and only very small readings are obtained using wire resistance strain gauges.

The method of measuring crankshaft stresses under running conditions is very reliable though difficult to set up and can only be applied to crankshafts with oilways in which to pass the wires to sliprings on the forward end of the engine. A system of telemetering a signal from a strain gauge and transmitter in the engine to a "wireless" receiver and recorder outside the engine has been developed with some success by B.I.C.E.R.A. and equipment will shortly be available from America which it is claimed is suitable for such applications.

An interesting point was made by Mr. Hinson on the measurement of torsional vibration in certain tug installations involving hydraulic couplings and reduction gears. Mr. Hinson suggests that the excitation of torsional vibration in these systems is related not only to the mean slip in the hydraulic coupling but also to the instantaneous variations in slip which must occur due to the nature of the applied torque and the load. Two methods could be applied to measure this, firstly, two self-generating vibration pick-ups could be mounted circumferentially one on each half of the coupling. An output from each, which would be proportional to changes in angular velocity, could be taken through sliprings to a recorder and the results compared. The second method requires the fitting of a toothed rim to the coupling. A magnetic transducer near the teeth would have a pulse generated by the proximity of each tooth and the voltage output would be proportional to the angular velocity of the rim. The rim would have to have a large number of teeth in order to resolve changes in angular velocity occurring in a short space of time.

Mr. F. ATKINSON

I would like to thank the Author for his most informative paper which I think fully illustrates some of the unusual problems which beset a Surveyor from time to time.

I would like to make one or two comments regarding the Author's remarks on "Eccentric rotation and malalignment problems". In particular, I wish to discuss Fig. 26 which I think does not fully explain the deflection pattern of a ship under various conditions of loading, and tends to confuse the reader. Firstly, it should be understood that the curves drawn in Fig. 26 only represent *change* between one condition and another, and are not to be mistaken for actual hull deflections. For one of the vessels investigated, a full longitudinal strength calculation was carried out and by successive integration of the weight curve for the ballast and coal cargo conditions, the hull deflection curves shown in Fig. "a" were plotted. These can be seen to be smooth, normal, regular curves with no "kinks". Also, it can be seen from Fig. "b" that the curves shown in Fig. 26 can be produced from these hull deflection curves by first plotting the ballast and load curves over the range A—B (which is roughly from the aft peak bulkhead to one third the length of the engine room from the after bulkhead) on a common base line, and then subsequently plotting a curve of difference, i.e. change. From the above, it can be concluded that the hull of the ship is behaving in a perfectly normal manner under load, although it does not appear obvious from the curves shown in Fig. 26 which were obtained from the investigation. Fig. 26 also suggests that points A and B are stationary, whereas Fig. "a" shows this to be erroneous. The Author states that one of the things roughly in common with the troublesome vessels investigated was the arrangement of deep tanks. The vessels had deep tanks both forward and aft which would hog the ships in the ballast condition. In the loaded condition the vessels would again hog and yet the initial alignment of the shafting and the chocking of the main engine took place when the vessel was in a light condition and sagging. I would like the Author's opinion on whether it would not be better to carry out the initial alignment when the vessel was hogging at some intermediate stage between the ballast and loaded condition, and not, as at present, when the vessel is bent to some line completely removed from an actual sailing condition.

With regard to the question of longitudinal or transverse framing, I cannot agree with the Author that a longitudinally framed ship will bend more than a transversely framed one; indeed, longitudinally framed ships have approximately 7 per cent more modulus than similar transversely framed ships. I should like to know whether the transversely framed ships investigated had midship deep tanks—also whether these were full at the time of investigation. If these were full the vessel would probably be sagging more than in the light condition and hence the initial alignment would have taken place on some line between the two sailing conditions.

However, I think the Author is correct in stating that the engine room double bottom stiffening should be carried further aft so that the No. 1 plummer block becomes integral with the main engine. This will decrease the amount of local deflection of the tank top caused by the relative change from light to heavy loading between the engine room and the hold immediately aft of the engine room. This stiffening should be gradually tapered off aft of the first plummer block to normal hold stiffening. However, it must be emphasised that this stiffening will only have a local effect and will not alter the overall hull deflection curves.

AUTHOR'S REPLY

To Mr. Archer

Many thanks for your kind remarks—as many of our colleagues know, you and I have grown up in the Society together and on many occasions your theoretical approach on obscure investigations has been of immense value.

How often, when seeking an indication of the true cause of a failure, have I sifted through the theoretical, metallurgical and practical aspects, knowing full well the danger of chasing "red herrings"!

I was very interested in your ideas of the qualities most valuable in a man for investigation work and whilst agreeing, would like to enlarge on your No. 4, i.e. "Power to interpret such measurements in terms of a satisfactory solution". It so often happens that measurements are taken and one has no standards with which to make comparisons; it is here that the investigator has to rely on his past experience. In other words, experience is a major factor in this "power to interpret".

Your description of the investigation into a case of trouble with top and bottom end bearings of the L.P. connecting rod of a triple expansion engine is interesting. Although, as in this case, when an engine has been operating satisfactorily and then suddenly develops trouble, this usually points to something incorrect having been done immediately prior to the trouble.

With regard to your query regarding the bedplate vertical deflections, according to the Surveyor who took the records on this job, the amplitudes and frequencies varied with the seaway, and I quite agree with your theory that "bending" of the whole engine in a fore and aft direction most probably causes the fracture of the guide bolts, in which case a greater stretching length should prevent this trouble recurring, although it is *not* the correct remedy.

The example of the twin screw triple expansion steam-engined installation was included in this paper as it was thought that it showed dramatically that although everything may seem in order at a survey in port with the machinery at rest, an entirely different proposition could exist when the vessel is at sea. The engines in this case were not fully balanced and the condition, as you suggest, most probably developed over the years as a result of the presence of large rotating unbalanced couples.

With regard to Fig. 19, I still feel that for boilers of the size fitted in the war-time "Ocean Class" ships, 10 in. between top row tubes and bottom row main stays is low. I agree, as I stated in the text, that scaling up of the tubes, subsequent overheating and differential expansion caused the trouble experienced in this case.

With regard to eccentric rotation of shafting, your interpretation that "when loaded with a dense cargo, such as iron pyrites, concentrated near the centre line, this could have set up transverse bending moments in the double bottom which the transversely framed ships would be better able to resist" is, I think, correct, and it was for this reason that Figs. 25(a) and 25(b), showing the different framings, were included.

With regard to the "definite pattern of crankshaft failures involving the forward side web of No. 4 crank in certain 4-cylinder Doxford engines, especially those in aft end installations"—in the case of some of these failures the short intermediate shafting was fitted oversize for torsional vibration reasons and supported in Michell bearings, and subsequent wear down of the crankshaft relative to the intermediate shafting, produced heavy bending stresses in the No. 4 crank. Why the failure usually took place at the forward side web of the No. 4 crank is hard to explain, unless it be that the torsional stress at the forward journal is higher than that at the aft journal.

To Mr. Atkinson

It is clearly stated on Fig. 26 that "on the assumption that alignment is a straight line in the ballast condition the graphs show the *change* in alignment when the vessel is loaded" and the purpose of Fig. 26 was to show that the local deflection at the *centreline* (where our shafting is situated) in way of the hold aft of the engine room, is greater with longitudinal than transverse bottom framing—more especially so still, when loaded with a dense cargo like iron pyrites.

I note that Mr. Atkinson "does not agree that a longitudinally framed ship will bend more than a transversely framed one"—this assertion was never made in the paper. It is not a question of the overall longitudinal stiffness of the vessel but the amount of local transverse support given to the shafting from the vertical ship's sides with which we, as engineers, are concerned. No doubt Mr. Atkinson is quite correct when he concludes that the hull of the ship, as a whole, is behaving in a perfectly normal manner.

Regarding the question of initial alignment, obviously shafting should be lined up when the vessel is in some intermediate stage of loading between sea-going ballast and full load, not dead light as is usually the case. However, this would not, I imagine, fit in with the normal building programme and as such is commercially impracticable.

To Mr. Guthrie

Your remarks about the steel strip sagging into a catenary are quite correct—and to get back to your little rhyme "the force" mentioned in the first line was, as I said in the paper, kept constant by referring to strain meter readings taken from a strain gauge bonded to the strip.

Thus, the catenary was used as a datum from which movements of surrounding objects could be measured.

Vibration of the strip did not occur and had it done so we would have arranged a dash-pot damper to counter same.

With regard to the twin-screw steam-reciprocating engined installation—the vessel was classed, and as I pointed out in my reply to Mr. Archer, this case was included as it showed dramatically that although machinery may seem in order when surveyed at rest in port, it can be an entirely different proposition under working conditions at sea. Your remarks regarding the Superintendent and Chief Engineer are noted—to know what is wrong is one thing, but to get it put right is another!

I was very interested to read of your experiences of hydrogen fires—the second case, in which the Wagner boilers with separate uptakes were involved, sounds more like simultaneous loss of water in both boilers through failure of feed pumps.

With regard to rechocking and realigning machinery to cure vibration, the only possibility of vibration being caused through malalignment, coming to my mind at the moment, is in turbines and gearing.

The large amount of alignment investigation work done by the E.I.D. has, in the main, been to reduce bending stresses in crankshafts.

My own experience of the E.I.D. was that the Shipowner genuinely appreciated the services we rendered, although, as in other lines of business, there were the rare occasions when we failed to give complete satisfaction.

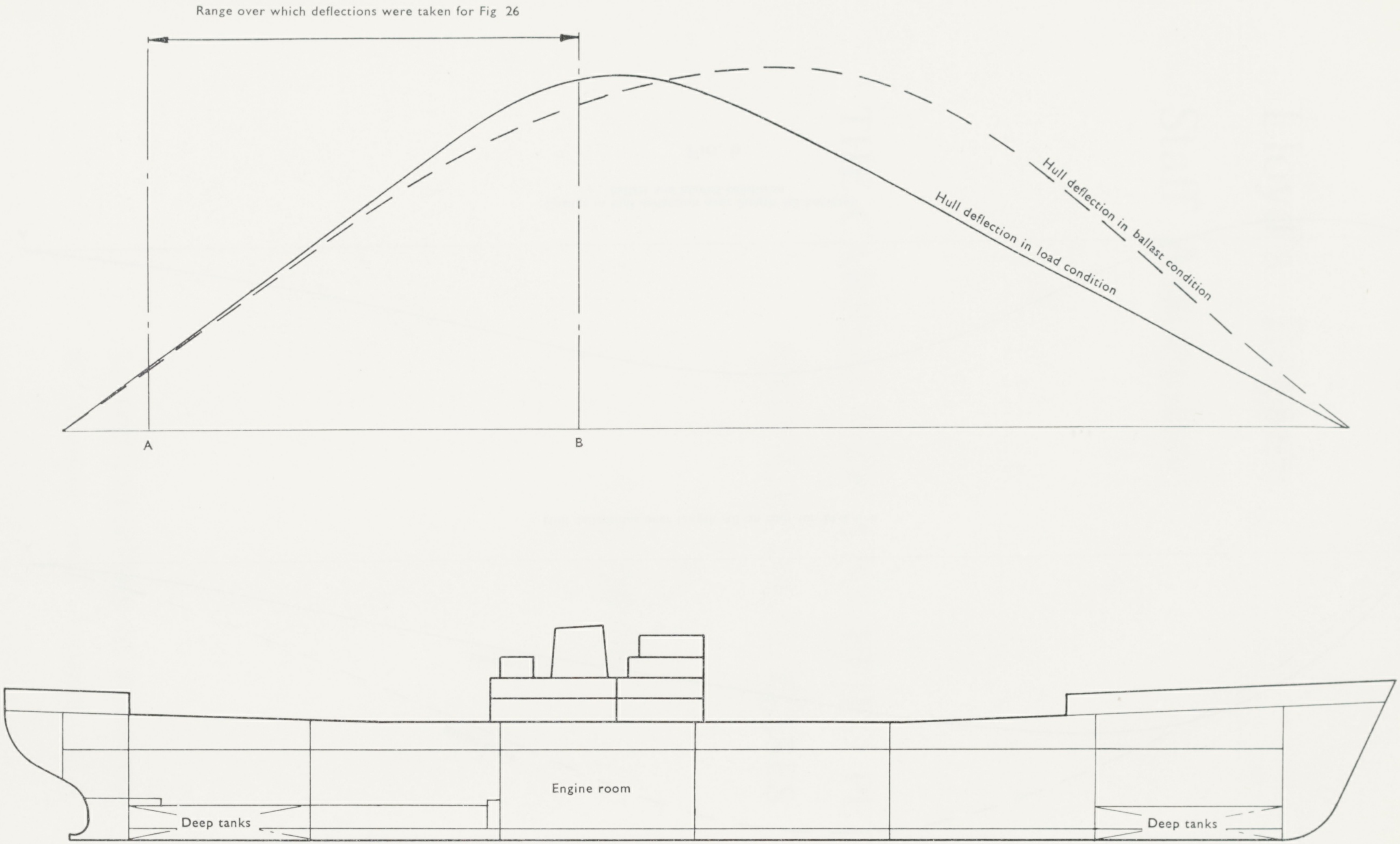
Range over which deflections were taken for Fig 26
Hull deflection in ballast condition
Hull deflection in load condition
A
B
Engine room
Deep tanks
Deep tanks

FIG. a

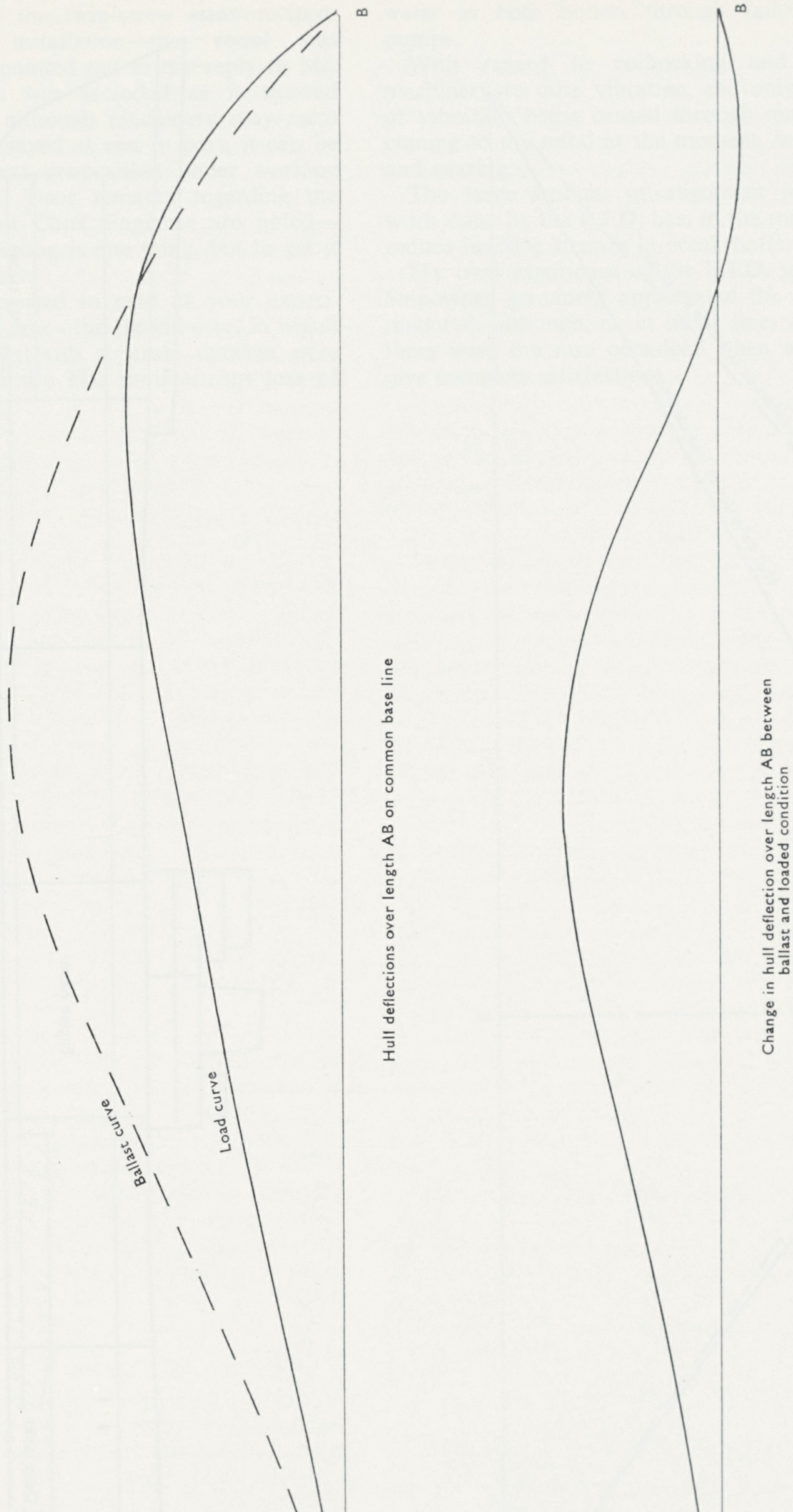
A
B
Ballast curve
Load curve
Hull deflections over length AB on common base line
A
B
Change in hull deflection over length AB between ballast and loaded condition

FIG. b

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THE CARRIAGE OF LIQUEFIED PETROLEUM AND NATURAL GASES

by

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LLOYD'S REGISTER OF SHIPPING

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The Carriage of Liquefied Petroleum and Natural Gases

By J. B. Davies

For many years some petroleum gases such as propane and butane have been carried by sea in pressure tanks at ambient temperature, but during recent years increasing attention has been given to the carriage of liquefied petroleum gases at ambient pressure, but low temperature. This interest has been brought about by the desire to transport methane since, as will be explained later, this cannot be carried under pressure as a commercial proposition, but must be transported at a low temperature. The investigations into the problems involved in this have also stimulated interest in the possibilities of carrying propane and butane at low temperature as opposed to what might be described as the traditional method of carrying them under pressure.

The Society recently issued Provisional Requirements for the Carriage of Liquefied Petroleum and Natural Gases at or near atmospheric pressure (Section D 70 of the 1962 Rules) and, in view of the developments which are taking place, these rules are couched in rather general terms and lay down standards to be complied with rather than specific and detailed requirements. The Author hopes that this paper will give his colleagues a general idea of the problems involved.

It might be as well to mention straight away that these problems are chiefly connected with the low temperatures at which the cargoes are carried. As regards an explosive risk, these cargoes are not considered to be any more dangerous than the petroleum products which have been carried in oil tankers for many years.

NATURAL AND PETROLEUM GASES

The natural gases, which are generally found in association with oilfields, consist chiefly of methane, but other hydrocarbons such as ethane, propane and butane are also present.

The term "liquefied petroleum gas", or L.P.G. as it is usually shortened to, is defined by the American Petroleum Institute as covering any material which is composed predominantly of any of the following hydrocarbons or mixtures of them; propane, propylene, butanes (normal butane or iso-butane) and butylenes. Of these the ones we are most concerned with are propane and butane.

L.P.G. may be obtained either from natural gas fields or from refinery gases when crude oil is processed, but in either case treatment is necessary to produce commercial gases. The commercial use of L.P.G. has been developed principally in the U.S.A. and in 1956 75 per cent of the production in that country came from natural gas and 25 per cent from refinery sources.

The use of the term "liquefied natural gas" is by custom reserved for the gas after those gases covered by the definition of L.P.G. have been extracted. It is thus principally methane and ethane.

Confusion is frequently caused by these definitions since a large proportion of the commercial liquefied petroleum gases is obtained from natural gas sources.

The composition of some typical natural gases is given in Table I, but it must be emphasised that these are given as a matter of general interest only, since the composition can vary appreciably between adjacent fields.

The Midlothian gas is included for interest only. It was discovered during trial borings by the Anglo-American Oil Company before the last war and has been used experimentally by the Scottish Gas Board.

Table II gives some properties of the principal gases with which we are concerned.

The critical temperature is that temperature above which the gas cannot be liquefied by pressure and it will be seen that for methane this is so low (−116° F.) that this method of carriage under pressure is, from practical considerations, not worth considering. Thus, while propane and butane can be carried under pressure at ambient temperature, methane must be carried at low temperature. When the liquefied gases are to be carried at atmospheric pressure the temperatures may be slightly lower than those given in Table II as the boiling points, since the commercial gases will never be completely pure, but will contain small quantities of other hydrocarbons having a lower boiling point. For design purposes −260° F. (methane) and −50° F. (propane) are normally adopted, although the Rules use −60° F. as a dividing point for some requirements since this temperature is necessary for certain grades of commercial propane.

CARRIAGE UNDER PRESSURE

Propane and butane may be carried under pressure since their critical temperature is considerably above ambient. The necessary pressure depends on the vapour pressure of the gas at the maximum temperature likely to be encountered in service (Fig. 1) and, to provide a reasonable margin, the tanks are usually designed for a pressure of about 250 p.s.i. unless it can be certain that commercial butane only will be carried, when a design pressure of 100 p.s.i. could be adopted.

TABLE I

SOME TYPICAL NATURAL GASES

	Sahara	Venezuela	Middle East (Typical)	Natural Gas (Texarkana)	Natural Gas (Cleveland)	Cousland Midlothian
	per cent					
Methane	81·3	90·9	72·9	96·0	80·5	84·4
Ethane	6·8	7·2		—	18·2	8·5
Propane	2·3	1·8	20·0	—	—	—
Butane	1·5	0·1		—	—	—
Pentane & Others	2·8	—	0·1	—	—	—
CO_2	0·5	—	4·0	0·8	—	0·5
N_2	4·8	—	3·0	3·2	1·3	7·0

TABLE II

PROPERTIES OF NATURAL AND PETROLEUM GASES

	Methane	Ethane	Propane*	Isobutane*	n–Butane
Molecular formula	CH_4	C_2H_6	C_3H_8	C_4H_{10}	C_4H_{10}
Boiling point at atmospheric pressure (°F)	–258·5	–127·5	–43·7	10·9	31·1
Critical temperature (°F)	–115·8	90·1	206·3	275·0	305·6
Explosive limits (% in air)—					
Lower	5·0	3·2	2·4	1·8	1·9
Upper	15·0	12·5	9·5	8·4	8·4
Specific gravity of liquid at B.P. ...	0·415	0·561	0·585	0·603	0·600
Ratio $\frac{\text{Gas volume}}{\text{Liquid volume}}$	585	406	295	229	232

* The figures quoted are for gases in the pure form. Commercial qualities for shipment will normally contain small quantities of other hydrocarbons, mainly ethane or isopentane and the properties will vary accordingly

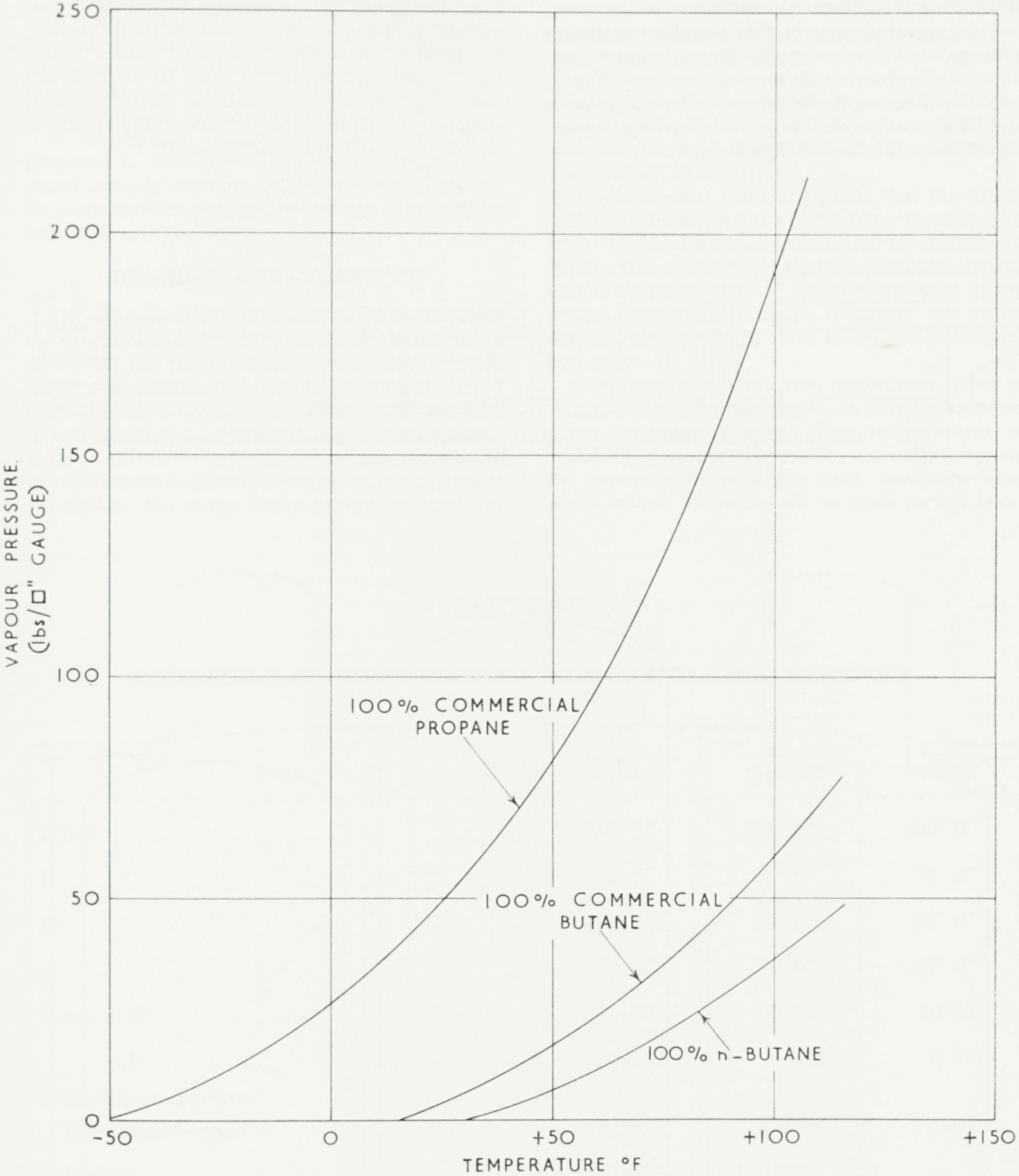


FIG. I VAPOUR PRESSURES OF PROPANE AND BUTANE.

To provide the desired flexibility in operation, however, the tanks are usually designed for the higher pressure.

The tanks are usually constructed of boiler quality steel in accordance with the requirements for either Class 1 or Class 2A welded pressure vessels. The method of determining the plate thicknesses is given in Appendix I.

The general arrangement of a typical small ship arranged for the carriage of propane under pressure in cylindrical tanks can be seen from Fig. 2, and Fig. 3 shows the midship section of the same ship. Thirty-six tanks are provided giving stowage for 52,000 cubic feet of liquefied gas.

CARRIAGE UNDER PRESSURE AT A TEMPERATURE BELOW AMBIENT

It will be evident from Fig. 1 that it is possible to reduce the design pressure if the temperature is reduced. While it might be thought that this introduced the troubles consequent on both pressure and insulation, an L.P.G. carrier has recently been completed for the carriage of liquefied commercial propane at 10° F. and 36 p.s.i. pressure. Presumably the savings associated with a reduction in pressure from 250 p.s.i. to 36 p.s.i. are considered to more than balance the provision of sufficient insulation and a refrigeration plant to enable the liquid to be carried at 10° F.

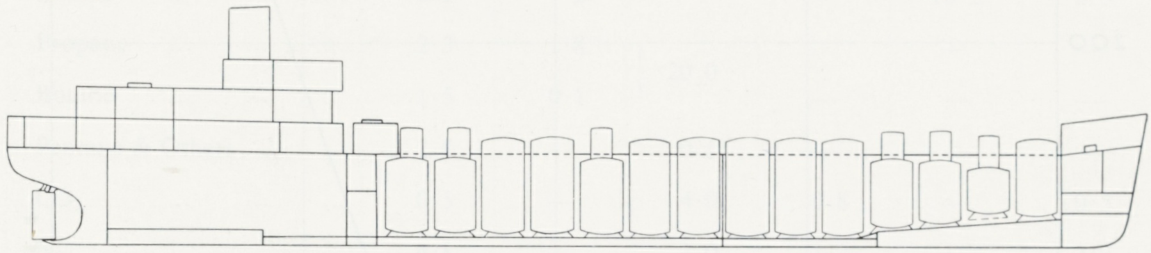


FIG. 2. GENERAL ARRANGEMENT OF VESSEL CARRYING PROPANE UNDER PRESSURE.

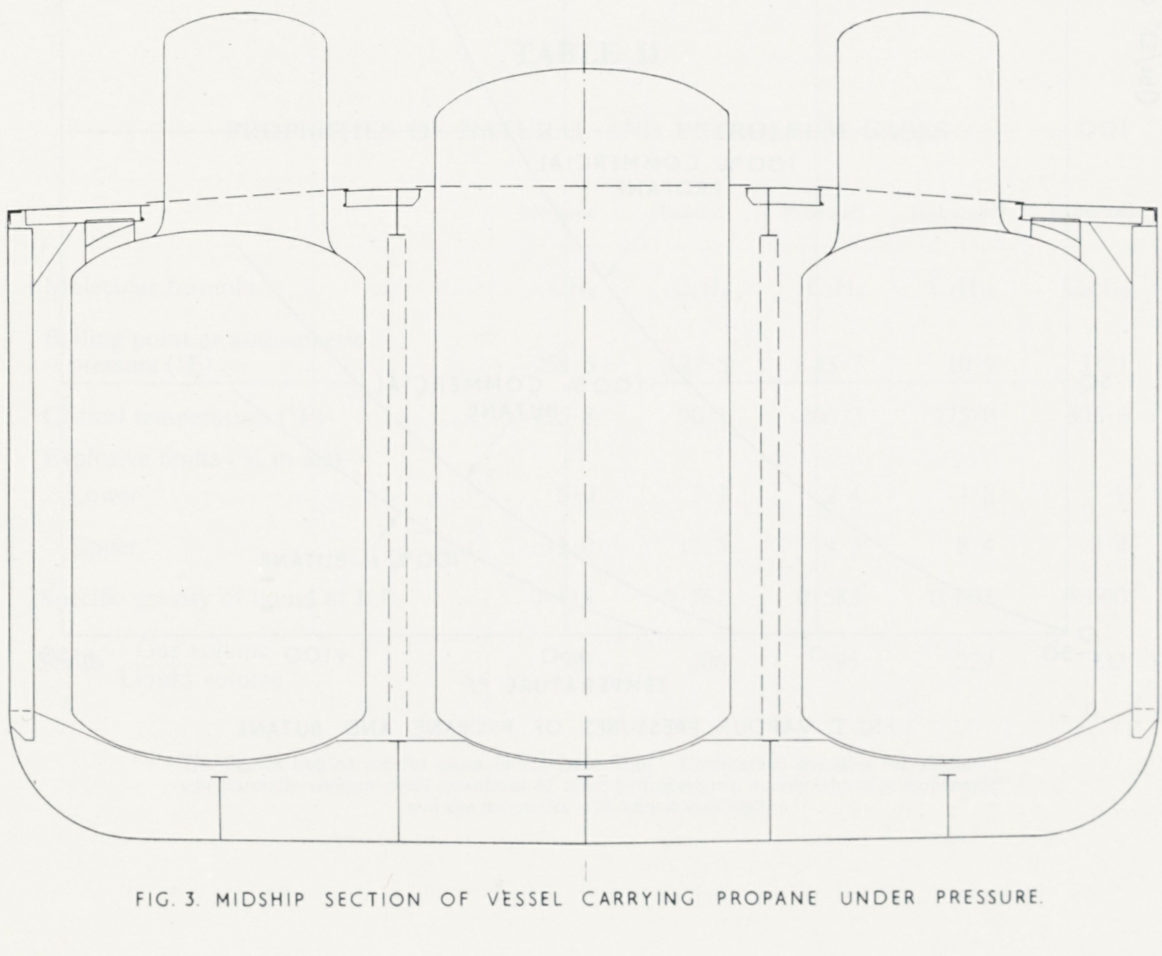


FIG. 3. MIDSHIP SECTION OF VESSEL CARRYING PROPANE UNDER PRESSURE.

CARRIAGE AT LOW TEMPERATURE

Undoubtedly the main interest to-day is in the carriage of methane at atmospheric pressure and low temperature. As remarked earlier the boiling point of methane is $-258°$ F. and $-260°$ F. is normally adopted as the design temperature.

It can be easily imagined that the necessity for such a low temperature gives rise to difficult problems in connection with the design of the tanks and their supports, the insulation and the selection of suitable materials. In addition, the lightness of the cargo raises general problems of hull design.

In the detailed discussion which follows, it is proposed to deal generally with the design for a vessel carrying methane, but where relaxations can be permitted for propane or butane, these will be indicated.

GENERAL ARRANGEMENT

The stowage rate of methane varies slightly with the source of supply, but it is generally about 80–85 cubic feet per ton or at least twice that of crude oil or most petroleum products. A methane carrier must therefore have a very large cubic capacity in relation to the deadweight. The design problem is made still more complicated by the necessity to provide separate spaces for water ballast as, unlike the tanker, the cargo tanks cannot be used for ballast on the return voyage because a certain amount of methane must be left in the tanks to keep them at a low temperature and avoid the necessity of spending a long time slowly cooling them before every loading. More space is needed to provide access round the tanks for inspection, so it will be obvious that the whole design revolves around the necessity for providing adequate cubic capacity. It can be expected that methane tankers will have a depth much greater than normal and a relatively low draught to depth ratio. While several design studies have been carried out, only a few have been published and the principal dimensions of three proposals are given in Table III.

It will be seen from this Table that the length to depth ratio is generally about ten compared with 13 to 14 for a normal tanker and the draught to depth ratio about 0·5, whereas a normal tanker would have a draught of about 80 per cent of the depth. These differences are, of course, due to the very different stowage rates of oil and the liquefied gases.

Cubic capacity is of great importance, since as much of the internal volume as possible must be given to cargo stowage. The temperatures at which these cargoes have to be carried necessitate the provision of relatively thick insulation and, while further reference will be made to this later,

TABLE III

DIMENSIONS OF SOME PROPOSED L.P.G. AND L.N.G. CARRIERS

	A	B	C
Cargo	L.P.G.	Methane	Methane
L.B.P. … … … … … …	574′ 2″	575′ 0″	600′ 0″
B … … … … … … …	82′ 0″	81′ 6″	78′ 0″
D … … … … … … …	54′ 9½″	58′ 6″	58′ 0″
d … … … … … … …	30′ 6″	26′ 0″	27′ 0″
Ratio L/D … … … … … …	10·48	9·83	10·35
d/D … … … … … …	0·557	0·445	0.466
Liquefied gas cargo:—			
Deadweight—tons … … … …	17,000*	12,000*	10,400
Barrels … … … … … …	180,000	170,000	—

References:—

A and B from "Low-temperature, liquefied-gas transportation" by Filstead & Banister, S.N.A.M.E. 1961

C from "Methane transportation by sea" by Corlett, R.I.N.A. 1960

* Derived figure—not in reference

the present practice is to keep the insulation from contact with the liquid. This means that the containment vessel will be at the cargo temperature and consequently very high temperature stresses would be set up if the containment vessel were an integral part of the hull structure.

Thus, separate cargo tanks are at present the normal arrangement and these are usually rectangular, since this gives the most favourable space utilisation factor. Suggestions have been made that they should be cylindrical and while this offers advantages from the strength aspect, it is wasteful of space.

If the insulation were fitted directly on the exterior of the cargo tank, it would be subjected to high stresses when the tank contracts or expands with change in temperature. As will be seen later, the contraction when the tank is filled can be quite large, and difficulty might be experienced in providing an insulating material which could stand up to these movements. The insulation is therefore generally arranged on the side of the ship's structure adjacent to the tank.

Whenever a general arrangement is being considered, it must be remembered that accidents can happen, and all reasonable design precautions must be taken to mitigate the effects of any failure which can be foreseen as being possible under normal service conditions.

Here the obvious point to consider is a fracture in the main cargo tank releasing liquefied gas at a low temperature into the surrounding space. This must not penetrate the insulation, and thus cause cold spots on the main hull structure, so a secondary barrier for the temporary containment of the liquid has to be provided. By "temporary containment" is meant a barrier which will remain efficient for the length of time necessary for the emergency discharge of the contents of the fractured tank.

It is also possible that there might be a failure in the insulation and, while this might be only over an extremely small area, it must be remembered that the inert gas in the space round the cargo tank will be at the low temperature of the cargo. Consequently, any breakdown in the insulation would lead to the formation of a cold spot on the adjacent hull structure. The provision of thermo-couples to detect any such spot will be discussed later ; it is sufficient at the moment to remark that, should this occur, arrangements must be made to circulate water immediately so as to prevent the steel cooling to a temperature such that brittle fracture would be likely.

This is normally arranged by providing a double skin construction at the side which forms a tank which can be flooded immediately if necessary. This space is also necessary as a water ballast tank to provide sinkage on the return voyage.

Transverse bulkheads will have the cold spaces on both sides and so, for all practicable thicknesses of insulation, the temperature of the steel would eventually approach that of the cargo although a small area at the edges would be at a higher temperature due to heat inflow from the sea water and the air. Obviously, brittle fracture problems would soon arise if special precautions were not taken. When the cargo temperature is not below $-60°$ F. (i.e. for propane) it is permissible to provide a normal single bulkhead but this has to be of Grade E steel and adequately insulated. For methane, however, this solution is impracticable due to the very low temperature, and for this cargo it is necessary to provide double transverse bulkheads (which may be of normal ship steel) arranged as floodable cofferdams. These can also be utilised as ballast spaces and flooding these with sea water on the return voyage, combined with the insulation on the tank side, will ensure that the temperature of the steel remains at an acceptable level.

Cofferdams are also required at each end of the tank space but, at the fore end, a water ballast tank could be accepted instead.

Many of the points mentioned above will be discussed in greater detail later in the paper, but we are now in a position to look at a typical general arrangement (Fig. 4) and midship section (Fig. 5) in which the features already discussed are shown in outline.

HULL STEEL

The grades of steel used in the bottom and side shell and in the sheerstrake and deck are governed by the Tanker Rules, but special consideration has to be given to the requirements for the longitudinal bulkheads, inner bottom and transverse bulkheads. It is these items which would be subjected to the low temperature if there should be any breakdown in the insulation and therefore, while the normal grade of steel can be used, certain special precautions have to be taken, or they must be of a grade of steel which would have satisfactory properties at this temperature.

These "special precautions" are designed to ensure that if any insulation breakdown should occur or if, in advance of an actual breakdown, there should be an unusual risk of one occurring, then immediate notice of this would be given so that the temperature of the steel could be maintained at a reasonable level by immediately flooding the adjacent water ballast space.

To give warning of a breakdown in the insulation, thermo-couples must be fitted on the longitudinal and transverse bulkheads and on the inner bottom so positioned that no part of these structural items is more than 10 ft. from a thermocouple. For a ship about 600 ft. long and of the general design shown in Fig. 5, about 300 thermocouples would have to be provided so it will be seen that the system has to be fairly extensive. It must also be so arranged that an audible warning is given as soon as a cold spot is found.

If the tank should fracture so that the liquefied gas escaped from the tank and came into contact with the secondary barrier, it is obvious that emergency action would be needed. Immediate warning must be given that a leak has occurred, and therefore the rules require that a device must

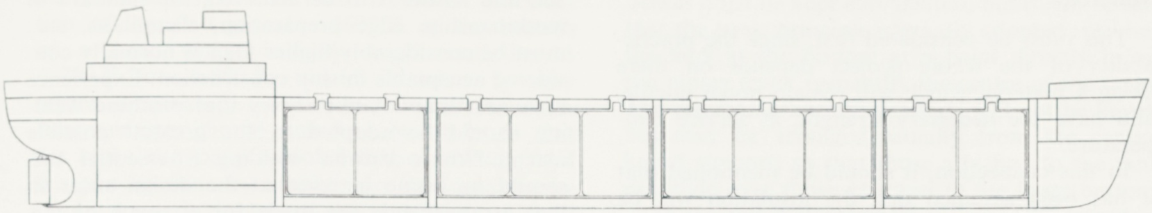


FIG. 4. GENERAL ARRANGEMENT OF VESSEL CARRYING METHANE AT LOW TEMPERATURE

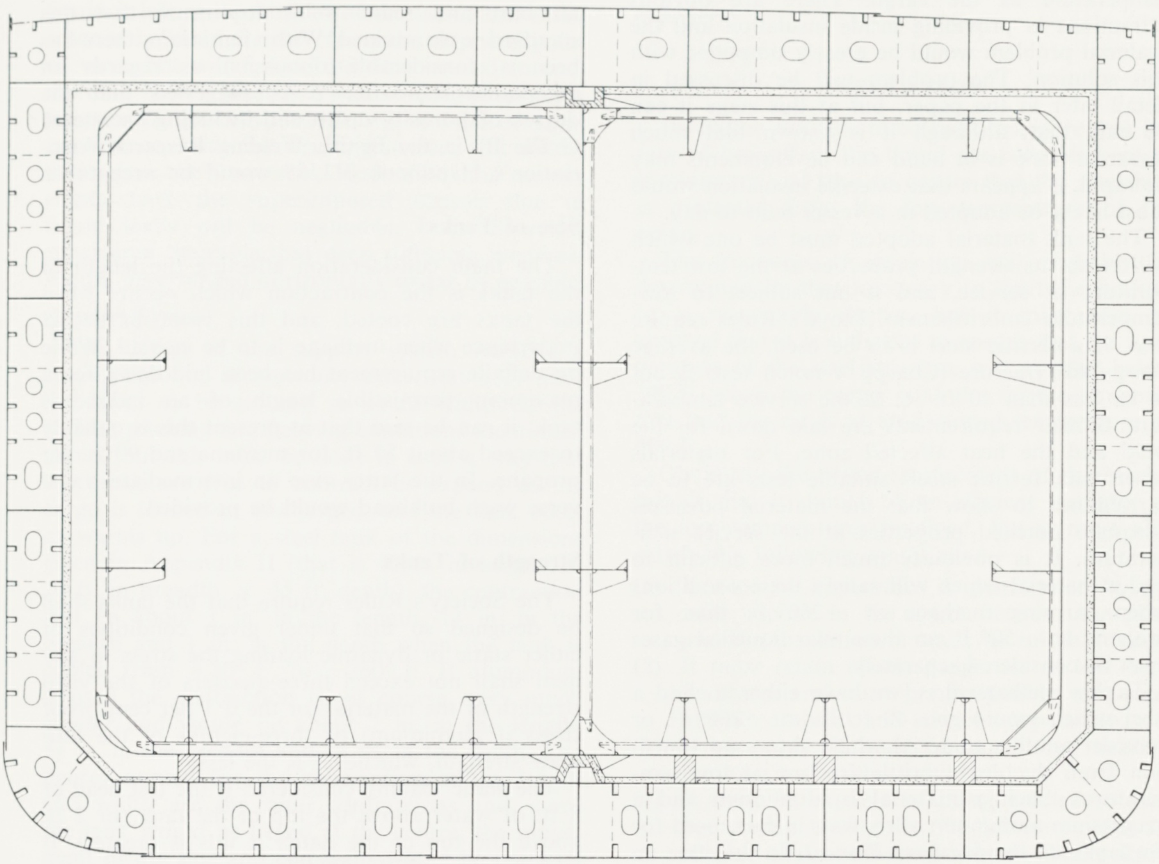


FIG. 5. MIDSHIP SECTION OF VESSEL CARRYING METHANE AT LOW TEMPERATURE.

be provided to sample continuously the vapour in the space between the tank and the insulation.

When the temperature of the cargo is not lower than −60° F. it is permissible to omit the thermocouples on the hull structure, provided the bulkheads, inner bottom and webs and transverses attached to them are made of Grade E steel.

LONGITUDINAL STRENGTH

The Society's Rules do not make any specific reference to the longitudinal strength requirements for this class of ship, but it will generally be found satisfactory if the section modulus is made in accordance with the tanker minimum, i.e. $\frac{I}{y} = KBC_b \times 10^3$. The still-water stress in the loaded condition will almost always be very low, due to the relatively small deadweight and good longitudinal distribution, but the stress in the ballast condition must always be checked. The bending moment in the ballast condition will probably be higher than that in the loaded condition, but the adoption of the minimum tanker section modulus will normally ensure a satisfactory stress in either condition.

CARGO TANKS

The design of the cargo tanks is obviously the most severe problem as any cargo to be carried at −260° F. is bound to give rise to many unusual questions. The main points to be discussed are:—

(1) Materials.
(2) Size of tanks.
(3) Strength of tank under both static and dynamic loading.
(4) Support system.

Materials

This must be considered first, since the practicability of the whole project depends on there being a material which will remain notch ductile, and have the necessary strength, at service temperature.

In this connection, it should be mentioned that it has been assumed that the insulation will be fitted outside the tank (normally on the hull structure) and consequently the tank is at the same temperature as the cargo. There are obvious attractions in providing inside insulation and the material problem would be greatly simplified with this solution. The problem will be discussed in detail later in the paper, but at this stage it can be said that, although it is known that much research work is in hand and developments may be rapid, it appears that external insulation would most likely be adopted in a vessel built to-day.

The tank material adopted must be one which will retain its strength properties at the low temperature of service, and is not subject to low-temperature embrittlement. Lloyd's Rules require that, if a ferritic steel is to be used, the average energy for fracture (Charpy V-notch test) is not to be less than 40 ft. lb. at the service temperature. Similar requirements are laid down for the weld and the heat affected zone. For materials other than ferritic steels suitable tests are to be carried out to show that the material possesses adequate notched properties at the service temperature. It is obviously much more difficult to find a material which will satisfy these conditions when carrying methane at $-260°$ F. than for propane at $-50°$ F. so these two liquefied gases must be considered separately.

Taking methane first, we have either to find a steel which would give 40 ft. lb. at $-260°$ F. or consider another material which shows no transition from ductile to brittle fracture at low temperatures. Such a material is aluminium and a magnesium-aluminium alloy was, in fact, used for the tanks on the *Methane Pioneer*. In this case an American alloy was used, but tests show that the British alloy NP5/6 would be equally suitable, i.e., in general, a $3\frac{1}{2}$–$5\frac{1}{2}$ per cent magnesium alloy.

Since the *Methane Pioneer* was completed a lot of work has been done in an endeavour to find a suitable steel, and tests show that a 9 per cent nickel-alloy should give the required impact values.

For propane the requirement becomes 40 ft. lb. at $-50°$ F. This is, of course, a much easier requirement to satisfy although it should be noted that this is still considerably severer than the requirements for Grade E steel, which call for 45 ft. lb. at 14° F. Aluminium would be acceptable, but there is really no need to go to this material since the requirements could be met by a $2\frac{1}{4}$ per cent nickel-steel or a chromium-copper-nickel-aluminium alloy steel such as A.S.T.M.-A410.

Whichever material is adopted, experience shows that very great care has to be given to the welding. A high standard of acceptance must be set, and if this is to be achieved, the standard of workmanship, edge preparation, cleanliness, etc., must be considerably higher than is normally considered acceptable in ship construction. Experience with aluminium tanks shows that machine welding should be adopted to the greatest possible extent. Where manual welding is essential this should be borne in mind at the design stage so that arrangements are made for as much of this as possible being downhand.

Radiography must be extensively employed on all butt and seam welds to ensure that this standard is maintained. With aluminium, there has been a considerable discussion as regards an acceptable standard; it is suggested that the degree of porosity up to and including "scattered Grade B" in the British Welding Research Association's Handbook H3/58 would be acceptable.

Size of Tanks

The main consideration affecting the length of the tanks is the contraction which occurs when the tanks are cooled, and this is of particular importance when methane is to be carried. While no definite requirement has been laid down for a maximum permissible length of an individual tank, it can be said that at present this is unlikely to exceed about 50 ft. for methane and 90 ft. for propane. In the latter case an intermediate transverse wash bulkhead would be provided.

Strength of Tanks

The Society's Rules require that the tanks shall be designed so that under given conditions of either static or dynamic loading the stress in any item shall not exceed three-quarters of the yield strength of the material (or the 0·1 per cent proof stress if aluminium) or three-eighths of the ultimate strength, whichever is the less.

The static loading considered is the test head of 8 ft. of water above the top of the tank (or 2 ft. above the top of the hatch if this is greater). It may be argued that this is very severe as the specific gravity of the liquefied gas is less than 0·6, but it must be remembered that the permitted design stress is relatively high compared with stresses normally worked to in ship design.

For dynamic loading it is necessary to take account of the combined effect of internal vapour pressure (normally 2 or 3 lb./sq. in.) and

(1) A complete roll of 30° port and starboard (i.e. through 120°) in ten seconds.
(2) A pitch of 6° half amplitude (i.e. through 24°) in seven seconds.
(3) A heave of L/80 ft. half amplitude (i.e. through L/20 ft.) in eight seconds.

Again, it may be said that it is very severe to require the tank to withstand all these simultaneously, but the same remarks regarding the design stress apply.

A typical calculation is given in Appendix II.

A point regarding the strength of the tanks which has not been mentioned is the possibility of high thermal stresses arising when the tanks are

being filled. It is evident that the sudden introduction of liquefied gas at $-260°$ F. at the bottom of a tank which initially might well be at $+70°$ F. or $+80°$ F. would introduce a very severe temperature gradient in the tank structure. Arrangements should therefore be made so that the tanks can be pre-cooled before loading, and this can be done by arranging a liquid gas spray system towards the top of the tank.

A vertical temperature gradient can also occur during the return voyage when a small amount of liquid is left at the bottom of the tank ; this small amount is "dead freight" from the shippers' point of view, but will reduce the time needed to cool the tank before loading. There will, however, be a temperature gradient, which will not be of great importance, since the resulting stresses will not by themselves be excessive, and with the nearly empty tank the superimposed stresses due to liquid loads will be negligible. It is, however, considered desirable that deep stiffening members should be arranged horizontally so far as possible.

Support System

The design of the support system requires a certain amount of ingenuity since, not only have the tanks to be secured so as to withstand the rolling, pitching and heaving conditions mentioned above, but this must be done in such a way that the tanks are free to contract or expand as the cargo is loaded or discharged, and the tank cools or warms up. For a steel tank of the dimensions given in Appendix II (that is 40 ft. in length $\times$ 70 ft. in breadth $\times$ 45 ft. depth), the contraction will be about 1 in. in the length, $1\frac{3}{4}$ in. in the breadth and 1 in. in the depth when the tank is cooled from $+80°$ F. to $-260°$ F. If aluminium is used the contraction is, of course, very much greater and would be about 2 in. in the length, $3\frac{3}{4}$ in. in the breadth and $2\frac{1}{2}$ in. in the depth. The figures quoted for the contraction in the length and width are the total contractions and, since the tank would be keyed at the centre, the movement of the side and end walls would be one half of these figures.

When propane only is to be considered, the tank naturally does not contract so much since the lower temperature is $-50°$ F. instead of $-260°$ F., but nevertheless the contraction is still sufficiently severe to prohibit the adoption of any system of entirely fixed supports.

A further point to be considered is that the design of the supporting system must incorporate sufficient insulation to prevent cold spots appearing on the ship's structure. The general question of insulation will be discussed later ; it will be sufficient here to state that in way of the bottom supports the insulation used must be of a type which, with an adequate bearing area, can take the load of the tank plus cargo and that in way of the side and end supports, it must be able to withstand the rolling and pitching forces.

Many designs have been prepared to provide suitable support systems and a great deal of time and money spent on this important feature. It will be appreciated that at this stage of development details must be kept confidential, but it can be said that the basic principle normally adopted is to key the tank at the bottom at a point directly below the piping trunk (generally the centre of the tank). The remaining bottom supports will not be keyed, but may be arranged radially from the central keyed support, so that those attached to the tank can slide over those attached to the inner bottom.

At the top it is generally sufficient to provide a central support keyed vertically to permit movement in this direction during the temperature changes.

INSULATION

Insulation must be fitted both to ensure that the rate of boil-off of the liquid gas cargo is kept to a minimum and also to ensure that the steel hull is protected from the low temperature of the cargo.

The thickness of insulation will probably be determined by that necessary to limit the boil-off to a reasonable degree, but it must also be checked that it is sufficient to ensure that the temperature of the steel structure will not fall below 30° F. when the cargo is at the service temperature and the air is at an assumed temperature of 40° F.

Obviously the prime requirement of the insulation is that it should be an efficient insulator at the low temperatures involved, but it must also satisfy the following conditions:—

(1) It must be impervious to penetration by the cold vapour in the space between the tank and the insulation or must be suitably faced.
(2) It must retain elastic properties at low temperatures and must be compatible with steel at both ambient and service temperature.
(3) If necessary, it must be capable of bearing the load, e.g. at the supports.
(4) It must be capable of being erected so that the sealing medium at the joints remains an efficient seal at all temperatures. Where there are several layers, joints in the various layers must be staggered.
(5) If in panel form, the panels must not be so large that serious sheer stresses are set up in the adhesives.

Ideally, a sixth point should be added—it should cost next-to-nothing!

On the *Methane Pioneer* balsa wood was used and this is suitable as a load-bearing medium. Unfortunately, though, it is extremely expensive to install and make tight. Materials of the expanded plastic or polyurethane foam types show promise of being suitable, and there are, no doubt, several others.

The requirement for a secondary barrier which would contain the liquefied gas in the event of a fracture in the tank means that either the insulation must itself be impervious to the liquid or it must be covered with an impervious facing.

All the foregoing is based on the assumption that the insulation is provided on the hull structure, but it is evident that a system with the

insulation inside the cargo tanks would have very great attractions since the tank shell would not then be subjected to the cargo temperature, and there would be no problems associated with contraction when loaded. The requirements for the material of the tanks could be much less severe, and it might even be possible to arrange the tanks as an integral part of the hull structure. Before such an arrangement could be accepted, however, it would be necessary to produce an insulation which could act as both main and secondary barrier ; that is its face would be the main barrier but if this were pierced then the liquid must not penetrate sufficiently far into the insulation to cause a cold spot on the tank material. In addition, the insulation would have to be strong enough to act as a containment vessel if the main steel shell were to fracture. Research on these lines is very active and it may well be that before long, a suitable combination of materials will be available to permit of this internal insulation system for use with liquid propane if not with liquid methane.

BOIL-OFF

There is bound to be some input of heat into the cargo during the voyage so some boil-off can be expected. Figures have been published for the experimental voyages of the *Methane Pioneer* and it appears that an average of 0·3 per cent of the cargo volume per day can be expected. The methane boil-off can be allowed to escape to atmosphere via a vapour pipe led to the mast head, and this is quite safe since methane is lighter than air. As an alternative, the boil-off could be reliquefied and returned to the tank ; this would necessitate the provision of refrigerating plant, and whether this would be worth while would, no doubt, depend on the length of the voyage. For a short voyage, it might be more economic to allow the gas to escape than to incur the capital and running costs of the refrigerating plant, additional piping, etc.

When propane or butane is carried the problems involved in reliquefying are not so severe, and the boil-off would be reliquefied and returned to the tanks. Only the emergency relief valve would vent to the atmosphere.

Suggestions have been made that, when methane is carried, the boil-off might be used for fuel and, subject to stringent safety precautions, the Society would accept the burning of methane under boilers or its use in the main engines of motorships. With a cargo of 10,000 tons the boil-off would amount to 30–40 tons per day which should suffice for propulsion purposes. Methane has a higher calorific value than oil fuel. For both the boiler and the diesel engine a dual fuel system would be used and all manœuvring would be done on oil fuel only.

PUMPING ARRANGEMENTS

The normal (if such a word can be used for a project which is still in the experimental stage as regards detailed design and economics) arrangement for emptying the tanks appears to be the provision of deep-well centrifugal pumps with the motor mounted on deck and the pump near the bottom of the tank. These pumps are designed basically to lift the liquefied gas out of the tanks, and it would be expected that booster pumps would be provided ashore or on deck, although the former will probably be more economic as the ships will presumably operate between fixed terminals.

Stripping the tank would not be necessary when discharging every cargo, since normally a certain amount would be left in to maintain the low temperature during the ballast voyage.

An alternative to the deep-well pump would be a submerged motor-driven pump situated at the bottom of the tank. This has not as yet been approved by the Society, and it is evident that very strict precautions would have to be taken to ensure that this would never be operating in an explosive atmosphere. This is discussed further later in the paper (*see* Electrical Installation).

It is evident that the pumps, pipes, etc., will be subjected to the low temperature of the cargo, and their material must be selected with this in mind. Similarly, provision must be made for insulating the pipes, ensuring that they are provided with temperature isolation from the hull, and for contraction and expansion of the pipes.

It is strongly recommended that all pipes should enter the cargo tanks above the weather deck. This will greatly reduce the fire risk, particularly with methane where the gas will quickly rise.

It has been previously mentioned when discussing the general arrangement, that provision must be made for leakage from the cargo tank into the surrounding space, and it is therefore necessary for pumping arrangements to be provided in this space for the prompt removal of any liquefied gas which might leak from the cargo tank.

Pressure-vacuum valves have to be provided on the tanks and each tank must have a liquid level gauging device, together with high and low level alarms. The purpose of the gauging devices is obvious, but the pressure relief valves must be adequate for two purposes. In normal service they must discharge the vapours formed by neat leakage plus the vapour displaced when loading. Emergency pressure relief valves are also provided so that, should a fire occur on board, these would provide means of discharging the vapours formed during exposure of the tank walls to fire.

Vacuum-relief is generally only necessary during discharging, since the tank is normally under pressure. This operation will be discussed later (*see* Safety Precautions).

ELECTRICAL INSTALLATION

In general, this must comply with the requirements for oil tankers, since any of the gases concerned are explosive within certain concentrations in air. Reference has previously been made to the provision of thermo-couples on cargo tanks and the adjacent hull structure, and these must be certified "intrinsically safe".

Intrinsically safe apparatus must be so constructed that when installed and operated under the specified conditions, any electrical spark that may occur in normal working, either in the apparatus or in its associated circuit, is incapable of causing an ignition of the gas or vapour.

It has been mentioned that deep-well pumps with the motors on deck are at present the "normal" arrangement, and these were used on the *Methane Pioneer*, but an alternative which is favoured from cost considerations is the submerged motor-driven pump. With this arrangement the motor and the pump form a combined unit at the bottom of the tank but, of course, current carrying wires must pass through the cargo tank. This application could only be considered for methane carriers where there will always be a positive pressure in the tanks.

The basis behind this proposal is that whenever there is any possibility of power being supplied, the atmosphere in the tank will be either the liquefied gas or gas vapour and no air will be present. While this may generally be true, arrangements would have to be made so that it would be impossible for any air to be admitted, and this would involve the connection of the inert gas system to the pressure vacuum valve. There are other problems in connection with the insulating materials and their safety arrangements for this system which are as yet unsolved, but research is still proceeding. As yet, this arrangement has not been approved by the Society.

The electrical installation also includes a system for continuously reading the thermo-couples on the hull structure, and this must provide for an audible warning being given as soon as any predetermined variation from the standard temperature occurs. There are, of course, a large number of these gauges and a scanning device has to be arranged for the successive reading of groups of thermo-couples. Should a cold spot be indicated, the scanner will then be concentrated on those couples in the immediate vicinity of this position with occasional readings of the complete system to provide warning should a further breakdown occur elsewhere.

The monitoring system required to indicate any leakage of gas into the space between the tanks and the hull structure adds a further complication to the electrical system.

SAFETY PRECAUTIONS, ETC.

Most of the principal safety precautions have already been mentioned, but it may perhaps be desirable to summarise them briefly, and this will also provide an opportunity to mention some of the precautions which have to be taken in service, together with some for which there has not been an appropriate heading so far in the paper.

When the work on the insulation is completed and the tank installed, the intervening space will be filled with air which will contain some moisture vapour. If this remained when the tank was cooled, ice would be formed and this might impair the efficiency of the insulation. This space is therefore filled with an inert gas and maintained at a pressure slightly above atmospheric. The inert gas must, of course, be one which would not condense at the service temperature, and nitrogen is suitable when methane is carried. The use of inert gas in this space is, of course, an added safety precaution although there should not be any source of ignition in the space between the tank and the insulation.

Before loading the first cargo, or after a tank has been emptied for survey, the air should be displaced by an inert gas as this obviates the dangers inherent in an explosive mixture being present in the tank which could arise if a small amount of gas were pumped into an air-filled tank.

Cooling down the tank must take place gradually, and it is very desirable to provide a spray system at the top of the tank so that the cold liquid is not directed on one portion of the tank. It is also necessary to provide a series of thermocouples on the outside of at least one tank so that temperature measurements can be made and a satisfactory pre-cooling procedure established.

During loading there will be a relatively large amount of gas to vent from the tank, since not only has the inert gas to be displaced, but part of the cargo will gasify due to heat input in the pipe lines, turbulence, vaporising in the tanks, etc. The ship's vent line can therefore be connected to the shore system so that the vapour can be recondensed. It has already been mentioned that liquid-level gauging devices and high-level alarms must be provided for each tank.

As soon as loading commences the thermocouple system on the hull and the system for sampling the inert gas in the spaces between the tanks and the insulation would be put into operation. It has already been mentioned that the thermo-couples must be constantly scanned automatically so that immediate warning is given of any breakdown in the insulation. Should this occur, then the ballast tank in way of the failure must be flooded at once to prevent the temperature of the steel dropping to what might be a dangerously low degree.

A fracture in the cargo tanks would not be shown immediately by the thermo-couples, since the escaping liquid or gas would be contained by the secondary barrier. However, it would be shown by the gas monitoring system and, since the secondary barrier is only designed for temporary containment, it is considered that the correct procedure if such an emergency should arise, would be to pump the contents of the affected tank, and any liquid which might have escaped into the surrounding space, overboard as quickly as possible. A loss of part of the cargo is better than the possible loss of the ship.

Before discharge commences, the shore gas line should be connected to the vacuum valve on the cargo tank, so that as the cargo is discharged gas will enter the tank and thus ensure that it is kept at a positive pressure. While a certain amount of

gas will be generated during unloading, it will almost certainly be necessary to take some from the shore supply.

As a further precaution against a vacuum being formed in the tank, it would be good practice to arrange a low-pressure switch which would stop the pump motor should the shore gas supply fail and the pressure in the tank drop to, say, 0·5 p.s.i.

It will be apparent that cooling down the tank will be a long process and will be a time waster if it is to be done before loading each cargo. At present it seems to be generally accepted that a certain amount of liquid cargo should be left in the tank. While, in the case of methane, this may not be sufficient to ensure that the tank is maintained at the very low cargo temperature, the temperature will not rise greatly and it is probable that only a very short cooling-down period will be required.

When the tank is due for survey, as much of the liquid cargo as possible will be removed and, as the remainder boils off, inert gas can be added so as to avoid the risk of an explosive mixture being formed when most of the gas has boiled off. Finally, the inert gas would be displaced by air.

SURVEY REQUIREMENTS

While experience with land installations indicates that there are unlikely to be any severe corrosion problems in the carriage of these cargoes at low temperature, it has been considered prudent to require that the inside of the tanks should be subjected to an annual survey. This requirement may well be reconsidered as further experience is gained.

It would obviously be expensive if the insulation had to be removed from the hull at every Special Survey, and in any case this is considered to be unnecessary, at least at the beginning of the ship's life. The Rules therefore say that, providing the insulation is adhering satisfactorily and there have been no traces of cold spots in service, removal of insulation will not be required until the vessel is eight years old. It must, however, be considered during the design stage that removal will be required eventually and sufficient access space must be provided to allow room for this work.

TEMPERATURES

The Fahrenheit scale has been used throughout this paper. It is appreciated that many Surveyors will prefer the Centigrade scale and therefore Appendix III gives the Centigrade equivalents of all Fahrenheit temperatures mentioned in the paper.

CONCLUSION

It will be evident that in endeavouring to prepare a paper giving a general view of this subject, the Author has received considerable help from colleagues in other departments in the London Office. To them he must express his sincere thanks. The way in which the information has been used and presented is, however, his own responsibility and if he has misinterpreted any of this advice he hopes it will be put right in the discussion.

The Author would also thank Mr. A. G. Kershaw for his assistance in the preparation of the paper and, in particular, for carrying out the calculations in Appendix II.

Provisional Requirements for Scantlings and Construction of Tanks for the Carriage of Liquefied Petroleum Gases at Atmospheric Temperatures on board Ships

1. Tanks to be constructed in general accordance with the Rules for Welded Pressure Vessels Chapter J 18, 19, 20 and 22, where applicable and with the Rules for Materials, Chapter P, Section 3.

2. Tanks to be to Class 1 requirements, or, where the thickness is $1\frac{1}{2}$ in. or less and the proposed minimum specified U.T.S. of the material does not exceed 28 tons/sq. in., they may be made to Class 2A requirements.

3. The scantlings of the tanks are to be in accordance with the following formulæ and the tanks otherwise to comply with the requirements laid down in the above Rules so far as applicable.

Cylindrical & Spherical Vessels

The required minimum plate thicknesses are to be determined by the following formulæ:—

(a) *Shells*

$$\text{W.P.} = \frac{C \times f}{D} (T - 0{\cdot}06)$$

Where W.P. = the maximum working pressure in lb./sq. in., which in general, is not to be less than the vapour pressure of the liquefied gas at 45° C.

T = the thickness of the shell plate, in inches.

f = the nominal working stress, in lb./sq. in., to be taken as $\frac{1}{3{\cdot}5}$ of the minimum specified tensile strength (see footnote).

C = 1·8 for Class 1 cylindrical vessels.
1·6 for Class 2A cylindrical vessels.
3·6 for Class 1 spherical vessels.
3·2 for Class 2A spherical vessels.

D = internal diameter of the shell in inches.

(b) *Dished End Plates* (concave to pressure)

$$\text{W.P.} = \frac{8{\cdot}4\,(T - 0{\cdot}03) \times f \times h \times J}{D^2} \times \left[\frac{\frac{r}{R} + {\cdot}05}{\frac{r}{R} + {\cdot}15}\right]$$

Where W.P. = the maximum working pressure in lb./sq. in., which in general, is not to be less than the vapour pressure of the liquefied gas at 45° C.

T = thickness of end plate in inches, after dishing which in no case is to be less than the thickness of an unpierced cylindrical shell of the same diameter and material.

f = the nominal working stress in lb./sq. in., to be taken as $\frac{1}{3{\cdot}5}$ of the minimum specified tensile strength of the material in tons/sq. in. (see footnote).

D = outside diameter of end, in inches.

J = 1·00 for end plates without cross-seams.
0·90 for end plates of Class 1 vessels with cross-seams.
0·80 for end plates of Class 2A vessels, with welded cross-seams.

R = inside radius of dishing, in inches.

r = inside radius of corner, in inches.

R_0 = outside radius of dishing, in inches.

r_0 = outside radius of corner, in inches.

h = external height of dishing, in inches, measured from centre of corner radius, which is to be calculated as follows:—

$$h = R_0 - \sqrt{\left(R_0 - \frac{D}{2}\right)\left(R_0 + \frac{D}{2} - 2\,r_0\right)}$$

(FOOTNOTE.—The shell thicknesses obtained by the above formulæ should be sufficient to withstand the stresses resulting from internal pressure and static liquid head, but may not be adequate to withstand the local stresses due to the weight of the vessels and their contents, plus dynamic loading in a sea-way at the seating attachments. This aspect will require special consideration in relation to the size of the vessel and to the design of the seating, bearing in mind that the rule thickness of a sphere, without reference to these factors, is approximately half that of a cylinder of equal diameter.)

4. Compensation should be provided for openings cut in the shell or end plates in accordance with Chapter J 1608, 1609 and 1612 of the Rules. In the case of spherical shells, compensation is to be provided for all openings the diameters of which exceed five times the thickness of the shell plus $2\frac{3}{4}$ inches.

5. Where the material used in the construction of the tanks is not of a type covered in P 3 of the Rules, full details of the chemical and physical properties, also heat treatment, should be submitted. In general, such materials are only to be used in Class 1 vessels and by manufacturers experienced in their fabrication.

6. Where the size of the tanks prohibits furnace stress relieving, full details of the proposed heat treatment procedure should be submitted.

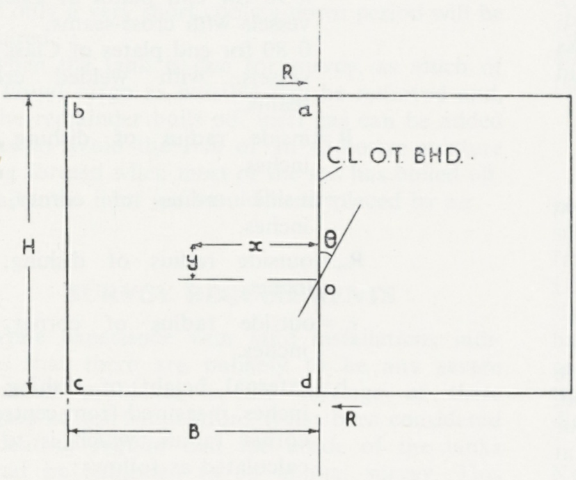
7. Suitable arrangements are to be made to prevent the pressure and temperature in the tanks from rising above safe limits. Details of the proposed arrangements are to be submitted, including number, size and type of relief valves, also sizes and diagrammatic arrangement of vapour discharge piping.

8. The support and securing of the tanks in the ship will require to be specially considered and full details are to be submitted of the support connections on the tanks and the supporting steelwork and the arrangements for permitting normal expansion and contraction of the tanks.

APPENDIX II

Methane Tank Scantlings

Rectangular Tank



R = Reaction at supports (Dynamic)

θ = Angle of roll

O = Ship's centre of roll

l = length of tank

1. Rolling. 30° p and s in 10 seconds

$$\text{Angular acceleration} = \frac{4\pi^2 \times 30 \times 2\pi}{10^2 \times 360}$$
$$= 0{\cdot}207 \text{ radians/sec.}^2$$

$$\text{Rotational couple} = 0{\cdot}207 \times \frac{I}{g}$$

(I = mass moment of inertia)

$$\therefore R \times H = 0{\cdot}207\,\frac{I}{g} \qquad \therefore R = \frac{0{\cdot}207\,I}{Hg} \text{ tons}$$

As the centre of the tank is not coincident with the centre of roll there will be translational components of acceleration.

i.e. Horizontal acceleration
$$= 0{\cdot}207 \times y \text{ ft./sec.}^2 = H_R$$
$$\text{Vertical acceleration} = 0{\cdot}207 \times x \text{ ft./sec.}^2 = V_R$$

2. Pitching. 6° half amplitude with pitch period 7 seconds.

$$\text{Angular acceleration} = \frac{4\pi^2 \times 6 \times 2\pi}{7^2 \times 360}$$
$$= 0{\cdot}084 \text{ radians/sec.}^2$$

Tank centre of gravity z ft. from amidships.

Vertical linear acceleration
$$= 0{\cdot}084 \times z \text{ ft./sec.}^2 = V_P$$

3. Heaving. L/80 ft. half amplitude in 8 seconds.

$$\text{Vertical linear acceleration} = \frac{L}{80} \times \frac{4\pi^2}{8^2} = V_H$$

Total vertical acceleration
$$= V_R + V_P + V_H = v \times g$$

Horizontal acceleration
$$= V_H \qquad = h \times g$$

The pressure inside the tank may now be calculated with the tank at the extremity of the roll and pitch.

Density of liquid = γ lb./cu.ft.

Relief valve setting = p p.s.i.

DYNAMIC LOADING

PRESSURE AT a

$$= p + \left[\frac{l}{2}\,\text{Sin } 6\,(1+v) + H \text{ Sin } 30 \times h\right]\frac{\gamma}{144}$$

PRESSURE AT b

$$= p + \left[\left(\frac{l}{2}\,\text{Sin } 6 + B \text{ Sin } 30\right)(1+v) + (B \text{ Cos } 30 + H \text{ Sin } 30) \times h\right]\frac{\gamma}{144}$$

PRESSURE AT c

$$=p+\left[\left(\frac{l}{2}\text{ Sin } 6+\text{H Cos } 30+\text{B Sin } 30\right)(1+v)\right.$$
$$\left.+\text{B Cos } 30\times h\right]\frac{\gamma}{144}$$

PRESSURE AT d

$$=p+\left[\left(\frac{l}{2}\text{ Sin } 6+\text{H Cos } 30\right)(1+v)\right]\frac{\gamma}{144}$$

STATIC (TEST) LOADING

Pressure at a and $b=8'$ water $=8\times\frac{64}{144}$ p.s.i.

Pressure at c and $d=(H+8)$ ft. water

$$=(H+8)\times\frac{64}{144}\text{ p.s.i.}$$

EXAMPLE—TANK SHOWN IN FIG. 5

$H=45'$ $B=35'$ $l=40'$

$L=580'$

$x=17{\cdot}5'$ $y=6'$ $z=165'$

$\gamma=26{\cdot}5$ lb./cu.ft. $p=2{\cdot}0$ p.s.i.

DYNAMIC LOADING

$H_R=0{\cdot}207\times6{\cdot}0=1{\cdot}24$ ft./sec.$^2={\cdot}039$ g

$V_R=0{\cdot}207\times17{\cdot}5=3{\cdot}62$ ft./sec.$^2={\cdot}112$ g

$V_P=0{\cdot}084\times165{\cdot}0=13{\cdot}86$ ft./sec.$^2={\cdot}430$ g

$V_H=\frac{580}{80}\times\frac{4\pi^2}{8^2}=4{\cdot}47$ ft./sec.$^2={\cdot}139$ g

$\therefore\ h=0{\cdot}039 \qquad v=0{\cdot}681$

PRESSURE AT a

$$2{\cdot}0+\left[20\times{\cdot}105\times1{\cdot}681+45\times{\cdot}500\times{\cdot}039\right]\frac{26{\cdot}5}{144}=2{\cdot}81\text{ p.s.i.}$$

PRESSURE AT b

$$2{\cdot}0+\left[(20\times{\cdot}105+35\times{\cdot}500)\,1{\cdot}681+(35\times{\cdot}866+45\times{\cdot}500)\,{\cdot}039\right]\frac{26{\cdot}5}{144}=8{\cdot}44\text{ p.s.i.}$$

PRESSURE AT c

$$2{\cdot}0+\left[(20\times{\cdot}105+45\times{\cdot}866+35\times{\cdot}500)\,1{\cdot}681+35\times{\cdot}866\times{\cdot}039\right]\frac{26{\cdot}5}{144}=20{\cdot}33\text{ p.s.i.}$$

PRESSURE AT d

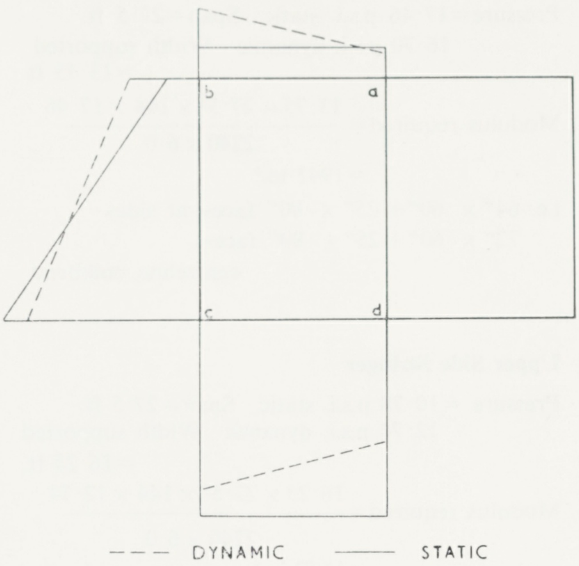
$$2{\cdot}0+\left[(20\times{\cdot}105+45\times{\cdot}866)\,1{\cdot}681\right]\frac{26{\cdot}5}{144}=14{\cdot}70\text{ p.s.i.}$$

TEST PRESSURES

a and $b=3{\cdot}56$ p.s.i.

c and $d=23{\cdot}56$ p.s.i.

Pressure Diagram



Determination of Scantlings

(a) Tank material aluminium alloy NP5/6

$0{\cdot}1$ proof stress $=8{\cdot}0$ tons/sq. in.

Ultimate stress $=17{\cdot}0$ tons/sq. in.

Design stress $=3/4\times8{\cdot}0=6{\cdot}0$ tons/sq. in.

Thickness of tank plating may be determined by applying a factor to the thickness derived from Table 32 of the Rules. This factor takes account of the properties of the material and the density of the cargo.

$$\text{Factor}=\frac{1}{2}\times\sqrt[3]{\frac{E_S}{E_A}}\times(1+\sqrt[3]{\gamma})$$

E_S and E_A are elastic moduli of steel and aluminium

γ = density of cargo

$$\text{i.e. Factor}=\frac{1}{2}\times\sqrt[3]{\frac{30\times10^6}{10\times10^6}}\times(1+\sqrt[3]{415})$$
$$=1{\cdot}259$$

Tank Plating—frames spaced 24 inches

	Head (ft.)	Table 32	Corrected
Bottom	45·0	·445″	·56″
Side Strake 1	42·5	·432″	·54″
,, ,, 2	35·8	·400″	·50″
,, ,, 3	29·2	·366″	·46″
,, ,, 4	22·5	·332″	·42″
,, ,, 5	15·9	·300″	·38″
,, ,, 6	9·2	·300″	·38″
Corner	2·5	·300″	·38″
Top	0	·300″	·38″

Lower Side Stringer

Pressure = 17·46 p.s.i. static   Span = 27·5 ft.
16·70 p.s.i. dynamic   Width supported = 13·75 ft.

$$\text{Modulus required} = \frac{13 \cdot 75 \times 27 \cdot 5^2 \times 144 \times 17 \cdot 46}{2240 \times 6 \cdot 0} = 1945 \text{ in.}^3$$

i.e. 64″ × ·60″ + 25″ × ·80″ face—at sides
72″ × ·60″ + 25″ × ·80″ face—centreline bulkhead

Upper Side Stringer

Pressure = 10·78 p.s.i. static   Span = 27·5 ft.
12·74 p.s.i. dynamic.   Width supported = 16·25 ft.

$$\text{Modulus required} = \frac{16 \cdot 25 \times 27 \cdot 5^2 \times 144 \times 12 \cdot 74}{2240 \times 6 \cdot 0} = 1677 \text{ in.}^3$$

i.e. 64″ × ·60″ + 21″ × ·80″ face—at sides
72″ × ·60″ + 21″ × ·80″ face—centreline bulkhead

Top Webs

Pressure = 3·56 p.s.i. static   Span = 32 ft.
7·23 p.s.i. dynamic   Width supported = 10 ft.

$$\text{Modulus required} = \frac{10 \cdot 0 \times 32 \cdot 0^2 \times 144 \times 7 \cdot 23}{2240 \times 6 \cdot 0} = 793 \text{ in.}^3$$

i.e. 42″ × ·60″ + 21″ × ·60″ face

Side Stiffeners

Centre Span

Pressure = 14·12 p.s.i. static   Span = 13·5 ft.
14·72 p.s.i. dynamic   Spacing = 24″

$$\text{Modulus required} = \frac{24 \cdot 0 \times 13 \cdot 5^2 \times 12 \times 14 \cdot 72}{2240 \times 6 \cdot 0} = 57 \cdot 5 \text{ in.}^3$$

i.e. 12″ × 6″ T

Top Stiffeners

Pressure = 3·56 p.s.i. static   Span = 9·25 ft.
6·67 p.s.i. dynamic   Spacing = 24″

$$\text{Modulus required} = \frac{24 \cdot 0 \times 9 \cdot 25^2 \times 12 \times 6 \cdot 67}{2240 \times 6 \cdot 0} = 12 \cdot 2 \text{ in.}^3$$

i.e. 6″ × 3½″ T

APPENDIX III

Centigrade equivalents of the principal Fahrenheit temperatures quoted in this paper.

°F.	°C. (Approx.)
−260	−162
−127·5	− 88,5
−116	− 82
− 60	− 51
− 50	− 45,5
− 43·7	− 42
0	− 18
10	− 12
14	− 10
30	− 1
31·1	− 0,5
40	4,5
70	21
80	26,7
90·1	32
206·3	97
275	135
305·6	152

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Discussion

on

Mr. J. B. Davies's Paper

THE CARRIAGE OF LIQUEFIED PETROLEUM AND NATURAL GASES

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

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Discussion on Mr. J. B. Davies's Paper

The Carriage of Liquefied Petroleum and Natural Gases

Mr. S. Archer

For some time there has been a need in Lloyd's Register for an authoritative account, not only of the physical considerations underlying the carriage of these hydrocarbon gases, but also of the Society's requirements so far as structural strength, materials, pumping arrangements, safety precautions, etc., are concerned. Personally, I cannot think of anyone better qualified to supply this need than Mr. Burton Davies.

As the Author points out, the carriage by sea of liquefied petroleum gases such as propane and butane, etc., under pressure at ambient temperature has been common practice for a number of years. Although the Society's Rules have so far contained no specific reference to such pressure tanks, it has been the practice to deal with them on the basis of Welded Pressure Vessels (Chapter J of current Rules) so far as scantlings, materials, etc., are concerned. Before long, however, it is intended that specific requirements, somewhat on the lines of the Author's Appendix I, will be included in the Rules. One of the changes envisaged is a reduction of up to 17 per cent in required shell thickness, depending on the class of vessel, i.e. Class IIa or Class I. This will have several advantages, including greater ease of welding and reduced stresses both static and dynamic, resulting from total tank weight.

The paper is so comprehensive and explicit that I find very little to comment upon. The Tables I and II with the graphs of Fig. 1 are very handy for reference and seem to cover most of the gases likely to be involved in this trade in the foreseeable future. One point not mentioned by the Author, however, is the filling ratio, or as the Americans term it, the "filling density". This is laid down by regulations and limits the weight of gas which may be loaded into a tank in order to avoid undue discharge from the relief valves as a consequence of thermal expansion of the liquid and its vapour due to ambient temperature changes or radiant heat effects. Where, as in gas cylinders, no relief valve is normally provided, this ratio becomes of even greater importance, of course! For most of the gases in question the limiting filling ratio, expressed as weight of gas as a percentage of the weight of water the tank will hold at 60° F., runs out at about 40–50 per cent. Slightly increased filling ratios are allowed where the tanks are lagged. As the Author points out, this represents a very light, bulky cargo compared with petroleum.

On the question of internal versus external insulation, would the Author agree that one of the possible hazards with the former may be its tendency to disintegrate, or become detached from the tank walls, under the dynamic action of liquid gas (i.e. "sloshing" effect) when the tank is only partially full?

I was rather struck by Fig. 3 in the paper showing the midship section of a ship carrying propane under pressure. If one has to reckon with three, or perhaps more, cylindrical tanks arranged abreast across the ship, does this not have a very considerable weakening effect and presumably it would be necessary to compensate by means of heavy doublings and/or longitudinals? Possibly, however, the lower loading due to the reduced density of L.P.G. compared with ordinary petroleum cargoes, plus the increased depth of hull girder vis-a-vis that of a normal tanker may make this effect more apparent than real? It would be interesting if the Author could expand a little on this subject.

One of the big problems with the pressure tanks, especially thin-walled, large diameter vessels, will be the support arrangements and this will clearly be more acute with the "compromise" system employing some refrigeration effect in order to lower the storage pressure. The resulting reduction in the shell thickness required to withstand gas pressure may well be excessive in the larger sizes of tank and may demand an over-riding minimum in the formula for plate thickness. In any case, the avoidance of points of stress concentration and adequate distribution of the support forces will be of great importance.

On the subject of "boil-off", rule requirements, including safety measures for burning under boilers, in diesel engines and in gas turbines, are at present in course of preparation and for this purpose consultation with the Ministry of Transport, the oil companies, boilermakers and other interested parties is proceeding. So far as low temperature methane is concerned, one of the main questions is whether it is economically more favourable to re-liquefy the "boil-off" by refrigeration and return it to the cargo or to use the vapour for propulsion purposes. This would depend, as the Author points out, upon the length of voyage, but of course another prime factor must be the price of methane gas in the consumer market. It has been estimated that to-day for certain routes reliquefaction would not become economic unless the market price of methane is at least £9 per ton. This is based upon an estimated overall cost of £4 per ton for refrigeration, the difference representing the cost of oil fuel which would have to take the place of the methane "boil-off". Would it be fair to say that at the present this market price is marginally competitive with coal gas, having due regard to the higher

calorific value of methane? It is as well to point out that the burning of "boil-off" from the heavier hydrocarbon gases, such as propane or butane, for propulsion purposes would not be permitted by the Society, because among other reasons, these gases are heavier than air and the resulting explosion hazard from possible leakage is therefore much greater.

Appendix II, giving typical calculations for determining the scantlings of methane tanks, including dynamic effects, is of much interest. Presumably, the assumed limiting values of roll, pitch and heave, which are the same as in D 70 of the Rules, could be regarded as exceptionally severe and their conjunction somewhat unlikely? On a small point of detail, it would seem by inference (although not stated) that the fluid pressures are calculated for the four corners, a, b, c and d, not at the ends of the tanks but at the mid-length of the tank. Presumably, for the relatively short lengths of tank envisaged, the slightly greater heads at the ends of the tanks due to pitching can be neglected?

MR. D. GRAY

I would like to thank the Author for a very interesting paper on a very topical subject but would like to ask the Author whether he could elucidate on one or two points.

1. A secondary barrier external to the tank insulation has been mentioned. In the paper this is described as for temporary containment only, but how long is it expected to hold up, i.e. will this barrier withstand any pressure?

2. The tank supports have been mentioned at some length and the need to design these so as to prevent heat loss and formation of cold spots. It must be mentioned, however, that the tank must be earthed in order to counteract formation of static charge.

3. Burning of the boil-off has also been mentioned. The Author has shown that gas from some sources contains as much as 5 per cent nitrogen. What would be the effect of this nitrogen when using the boil-off as fuel, and is it necessary to know what percentage of nitrogen is present?

This question is asked because it is not possible to measure the nitrogen by direct methods and to measure it by deduction would involve very complicated and expensive instrumentation.

4. What methods are envisaged for tank stripping?

5. How much liquid will be left in the tanks for the ballast voyage in order to keep temperature down?

6. If one tank were to lose its dead freight it would be necessary to fully cool-down this tank prior to arrival at the loading port. Has the Author any suggestion as to how the recirculation of remaining cargo could be carried out?

7. Intrinsically safe thermo-couples have been mentioned. It would be as well to note that, as yet, there are few countries which produce intrinsically safe equipment and there is no international specification yet for such equipment. Only the U.K., U.S.A. and Germany have any testing establishments for intrinsically safe equipment.

It is also important to realise that a certificate of intrinsic safety includes the circuit as well as the equipment and that, therefore, all cabling is included in the specification as well as the instrument.

8. Finally, the possibility of using a submerged electric pump has been described with the emphasis placed on reduced costs. This is by no means the only reason, and I would like to give my own views on the relative merits of submerged pumps versus deepwell pumps for such ships.

(*a*) *Position in tank*. Submerged pump is flexible and can be placed in the most advantageous position for drainage with minimum trimming of the ship.
The deepwell pump is inflexible being restricted to a position vertically below the trunk, i.e. centre of tank unit.

(*b*) *Driving unit* with the submerged pump is adjacent to the pump. With the deepwell pump it is remote from the impeller. A long drive shaft is necessary which will increase in length as ships increase in size.

(*c*) *Shaft seals* are needed with the deepwell pump only and can be a source of trouble.

(*d*) *Loading line* could be incorporated in the discharge line if a submerged pump were used. With a deepwell pump a separate line is required.

(*e*) *Overhaul of drive unit* with the submerged pump can only be done after gas-freeing, whereas with the deepwell pump some work could be done without gas-freeing.

(*f*) *Speed*. With the submerged pump relatively high speed is possible, whereas the deepwell pump is restricted to relatively low speed.

(*g*) *Materials*. With the submerged pump special materials are required for both motor and pump, whereas with the deepwell pump special materials are required for the pump unit only.

(*h*) *Cost*. Submerged pump could well be as low as 50 per cent of cost of deepwell pump, but the cost of safety precautions associated with the submerged pump may very well invalidate this saving.

If the submerged pump is ever adopted, stringent safety precautions will be necessary to maintain an oxygen-free atmosphere in the tank.

Regarding materials to be used, they would have to withstand the thermal shock during the cooling-down period, say 36 hours, and this would apply to:—

Insulating materials.
Mechanical strength materials.
Bearings.
Cable.

Finally, it would be wise to use a low voltage motor, e.g. 115 V rather than 440 V to minimise any corona effects.

Mr. R. G. LOCKHART

I would associate myself with the compliments paid to the Author for his presentation.

This contribution and any comments made can be taken as applicable to the natural or methane gas carrier.

When one considers the abnormal ratios of the dimensions, coupled with the small ratios of draught to depth and deadweight to cubic and the fact that the ship will make the return trip in ballast, the project does not appear at first sight, to be an exercise in economics.

Perhaps this is a short-term outlook when it is considered the Canvey Island project served by two methane tankers of about 170,000 bbl. would be capable of supplying about 15 per cent of the current domestic gas requirements of the U.K.

The Author echoes the remarks of earlier writers on the subject when he states that the explosion risk is considered to be no more dangerous than for general petroleum products.

It should, however, not be lost sight of that a land storage mishap occurred at Cleveland some 18 years ago.

The tank ruptured and in a remarkably short period the liquid methane had spread into the adjacent residential area, generally freezing everything in its path, eventually vaporising and exploding, causing 128 fatal casualties and tremendous damage to property.

The tank was constructed of $3\frac{1}{2}$ per cent nickel steel and failed in a brittle fracture. It is noticed in the section under materials a 9 per cent nickel-alloy steel should give the required impact values.

The protection required, and particularly the secondary barrier, makes it difficult to accept totally the suggestion that the risk of carrying methane is not higher than for carrying petroleum.

The Author states that the probable size of aluminium storage tanks for ship storage would be 40 ft. $\times$ 70 ft. $\times$ 45 ft. and the proportional contractions would be 2 in., $3\frac{3}{4}$ in. and $2\frac{1}{2}$ in. The extensive requirements for the use of radiography are readily understood. While an amount of machine welding can be used, a considerable proportion of the remainder will require to be done without machines and in positions other than downhand. The stresses at the boundaries, and particularly at the corners, could be expected to be of quite a high order and the construction and workmanship in these regions will require to be of the very best.

One cannot help feeling that the exploitation of fitting insulation inside the tank must be expedited.

The interest in transportation of methane by sea is growing in Japan and Western Europe.

The Germans have announced that they are experimenting with sheet Styropor in association with a thin alloy surface and, while they have not yet overcome contraction problems entirely, the experiments so far appear to be very satisfactory.

Referring to Fig. 5, in the event of a primary failure and liquid entering the secondary compartment, is it to be supposed that the voyage can continue or is it expected that the cargo will be jettisoned as soon as possible?

While it is appreciated Fig. 5 may not be to scale, it appears that the secondary compartment would not be sufficient to permit work being done in place in this region.

Taking a long-term view, presumably the outside of the tank and the inside of the cofferdam would require to be surveyed, and in the event of renewal being required, with the arrangement as shown in Fig. 5 it would be necessary to unship the tanks.

Mr. H. R. CLAYTON

This paper, which is mainly concerned with the general principles governing the bulk carriage of hydro-carbon gases at their atmospheric boiling points, will repay careful study, and if the principles are fully grasped, they will provide a good foundation for further knowledge on this interesting subject.

It is fairly certain that, as in the case of oil tankers, the design of gas tankers will settle down, and in the years to come it is possible that tremendous developments will take place in this type of ocean transport.

At the top of page 10 Mr. Davies refers to the advantages to be gained by having the insulation inside the cargo tanks, and I should like to ask him if he can tell us why the original idea of lining steel tanks with balsa wood was abandoned. It occurs to me that doubts may have been entertained as to the reliability of the glueing processes for securing the first layer of balsa wood to the tank and subsequently building up the thickness, but I am not sure on this point.

Under the section on pumping arrangements on page 10, the submerged motor driven pump is mentioned as a possible alternative to the deep well pump. These pumps, of course, would handle the bulk of the liquid in the tank. There are, however, two other interesting pumping devices which deserve some mention since they are useful for stand-by duty and for stripping purposes, viz. the blow-case pump and the vapour-lift pump.

The former consists of a small container situated at the bottom of the cargo tank which fills by gravity through a non-return valve and is discharged by gas pressure. The gas line serves alternately as a vent whilst the container is filling and as a high pressure gas supply to the container for discharging its contents. An automatically-operated three-way cycling valve on deck controls the gas supply to, and the exhaust from, the container. This is an inefficient but effective and reliable pumping device.

The vapour-lift device consists of an open-ended lift pipe in the cargo tank which is connected to a separator tank on deck. A partial vacuum is created in the separator tank by connecting it to the suction side of a gas compressor and this causes some of the liquid in the lift pipe to flash into vapour with the result that a liquid/vapour mixture flows up the lift pipe into

the separator tank. The liquid which collects in the separator tank is pumped ashore by a special pump.

I should like to draw attention to the paragraph at the end of the section on pumping arrangements which is not as happily worded as it might be. The impression could be gained that vacuum relief *is* generally necessary during discharge of the cargo, in spite of the fact that the sentence concludes by stating that the tanks are normally under pressure. In actual practice the pressure in the tanks is maintained between $1\frac{1}{2}$ lb. and $\frac{1}{2}$ lb. per sq. in. and the cargo pump would automatically stop if the pressure in the tank fell below the lower figure. A complete loss of pressure in the tank followed by the opening of the vacuum relief valve is regarded as a remote possibility.

The associated paragraph at the bottom of page 11 also needs a slight correction. The vacuum relief valves are not connected to the shore gas line, but open either to the atmosphere or the inert gas line. The short gas line is permanently connected to each cargo tank trunk.

In conclusion, I should like to thank the Author for a helpful and interesting paper.

Mr. C. D. SNEDDON

Referring to page 1, the Author gives the operating pressure for propane and butane pressure vessels at 250 and 100 lb./sq. in. respectively, which, judged from the graph Fig. 1, represents service temperatures of between about 120° F. and 130° F.

These seem somewhat on the low side considering an appreciable portion of the hull and the tanks themselves will be exposed to direct sun, unless a deck cooling sprinkler system is installed as a requirement associated with these pressures.

I am thinking of the comparable case of liquefied CO_2 in cylinders, when many years ago a gas cylinder committee investigating temperature limits compromised on a maximum operating temperature in the sun of 150° F. This was somewhat less than the sum of the maximum low altitude shade temperature of 136° F. recorded in tht Mediterranean area and 25° F. for exposure of steel to the sun, based on Panama Canal figures found to be between 25° and 40° above shade temperature.

Thus, the estimated temperature allowed for before a CO_2 bottle will blow off is about 30° higher than the apparent allowance for propane. It would seem there could be considerable boil off of cargo in warm climates, and perhaps the Author could enlarge on the point as to what considerations decided the choice of 250 lb. and 100 lb./sq. in. for petroleum gases.

My second point concerns general safety considerations. To say, as regards explosive risks, the carriage of L.P.G. and natural gases is no more dangerous than petroleum products that have been carried for years, savours of rather dubious complacency. Spontaneous ignition is not a risk to be considered as the necessary temperatures lie between 800° F. and 1,000° F., but a gas under pressure is itself a higher risk apart from its having a lowered flashpoint.

If the calorific value of the gases is above that of oil fuel, a blow-out of any pipe joint or valve could, if ignited, produce a white heat near the point of escape within minutes, with the probable disintegration of the structure locally, followed undoubtedly by an explosion. The results would be more catastrophic than any tanker explosion. Perhaps the Author could state whether, to meet such a contingency, nitrogen under pressure is led to the pressure containers to dilute the gas to below the flammability range, should this become necessary.

Regarding safety of life precautions generally, one danger that has been completely neglected in the design of gas tankers is that arising from collision. I don't think there is any comparison with ordinary tankers in the resultant destruction. For instance, tons of liquid methane could be thrown into contact with sea water having a temperature 300° F. or more above its boiling point. Neglecting fire and explosion there will be an immediate volcanic boil-off, and scattering of liquid methane rather like pouring water on to burning oil. About 45,000 cu. ft. of gas at −258° F. will be produced per ton of methane involved, the entire atmosphere surrounding the ship being subjected to a drastic reduction in temperature that could freeze or cold-burn anyone to death within a very short space of time. The air itself will be unbreathable due to its temperature and asphyxiating effects, a 10 per cent concentration of methane being fatal if breathed for any length of time.

Any chance of launching lifeboats may well disappear in ice formations, and covered inflatable liferafts would be petrified by contact with the liquid. Both ships in the collision would be involved in unprecedented difficulties, whether or not brittle fracturing of the hulls occurred from reduced temperatures.

There appears here a case for serious reappraisal of safety of life precautions, and if it cannot be made compulsory, for economic reasons, to keep the gas tanks inboard of the B/5 convention collision penetration limit, then I consider every gas tanker should be provided with one or more insulated, air conditioned safety citadels to which the crew could retreat until conditions outside had ameliorated sufficiently for abandoning the ship.

If the Author knows of any collision cases involving gas tankers, it would be of interest to know of the consequences to both ships concerned.

I would also like to ask the Author whether he can give any indication of the beginnings of a code of safe practice for gas tankers, such as is contemplated by Recommendation No. 15 of the 1960 SOLAS Convention. This Recommendation calls for the preparation of internationally agreed regulations covering the safety measures to be

taken in the construction and operation of tankers, to replace those of individual Governments and Port authorities—having in mind the fire and explosion risks particularly.

Precautions to be taken in the storage and movement of petroleum products on land and in ports are governed by the Petroleum Consolidation Act 1928—but statutory precautions covering carriage at sea appear to be non-existent.

The M.O.T. appear satisfied if the statutory fire appliances are provided, and apparently owners and masters simply apply what is considered "good practice" in regard to safety precautions at sea.

This will probably be altered by the 1960 Convention which requires the issue of Cargo Ship Safety Construction Certificates, certifying the ship is in all respects satisfactory for the service intended. If *satisfactory* means satisfactory out of harbour as well as in, then it would seem a code of safe practice for sea carriage is equally necessary as one for land and ports.

If the Author has any information on this it would be welcomed.

In conclusion, I would like to add that the Author is to be congratulated on a most interesting survey in a developing field of sea transport.

Mr. G. M. BOYD

This is a valuable paper, which widens our knowledge of a topical problem.

The definition of "LPG" is particularly welcome, since it is sometimes thought to mean "Low Pressure Gas". It is a pity, however, that the Author did not adopt Centigrade and give Fahrenheit equivalents instead of the other way about.

Referring to page 6, bottom of left column, is there not a danger that the water admitted to a cold space may freeze and so become a less effective heat supply?

Regarding material (page 8), this is one of the most important and difficult problems, which is by no means solved. The risk of disaster due to material failure by brittle fracture is a vital consideration, as pointed out by other speakers who referred to the Cleveland disaster. Mr. Sneddon's point about collision risk is also very apt. The Author mentions materials "which are not subject to low temperature embrittlement". This, of course, is an abbreviation of what is really meant. All materials become brittle if the temperature is low enough. The problem is to select a material which is not unduly brittle at the operating temperature. So far, the only steel which can be considered for liquid methane is the austenitic type, which is very expensive. Aluminium alloys, and only some of these, are the only practical solution for such a low temperature.

Referring to the formula for working stress (page 8, right column), this adds another to the many such formulæ that have been proposed, and it seems time that some uniformity were sought.

Referring to the section on boil-off (page 10) it may be of interest to know that for many years the methane evolved in sewage disposal has been used as fuel for diesel engines driving pumping equipment at sewage disposal works in this country. The dual-fuel diesel engine was developed in this country for this purpose, and it is fortunate that such engines are now available to meet the problem outlined in the paper.

On page 11, safety precautions against explosion are mentioned. It could perhaps be emphasised that the danger of explosion arises from the admission of air to the tanks. If air is excluded there is practically no risk. If an explosive mixture of air and gas exists, one of the worst risks is static electricity, so that the spray system mentioned would have to be efficiently earthed. This point has been touched upon by Mr. Gray.

Mr. O. M. CLEMMETSEN

Although I have been associated with the examination of the hull scantlings of the many proposals for L.P.G. carriers which have been received in the Ship Research Department (very few of which, incidentally, seem to come to fruition), I have found this paper very helpful in clarifying the subject as it sets down much more clearly than is possible in the Provisional Requirements, the various factors involved in the structural design of this type of ship. The Provisional Requirements can only point out the Society's approach to those who are trying to solve problems with which they are already familiar, but in this paper the problems themselves are presented.

The method of determining the tank scantlings is fully explained in the Appendix and in the body of the paper it is mentioned that the minimum modulus is to be that of a tanker, but some remarks regarding the local strength may be of interest. In view of the low L/D and small d/D, it would obviously penalise such ships if the strength of individual items were also based on tanker standards. The shell plating therefore is at present approved on a dry cargo ship basis, and if the thickness of the deck plating is based on the rule minimum for a cargo ship, it is usually found that the tanker minimum modulus is obtained. The thickness of the inner deck in a design such as Fig. 5 should also bear a relationship to a minimum second deck thickness as well as to the thickness necessary to withstand the hydrostatic pressure.

The strength of the stiffeners and plating of the inner longitudinal bulkheads is based on deep tank standards where ballast is to be carried in these spaces. In proposals where the double bottom and side tanks are common from deck to centreline, the tank top plating must be suitable for the increased head compared with a normal dry cargo ship.

Spaces such as transverse cofferdams are also based on internal water pressure, but the bulkhead as a whole, i.e. webs and stringers, must be equivalent to watertight bulkhead requirements for a pressure from one side of the bulkhead.

Side shell longitudinals are usually based on tanker requirements without the D/6 addition to the head, attention being paid to the strength of the uppermost longitudinals in relation to the test head. Side shell webs are also based on tanker standard with a reduction to take account of the reduced draught.

Having regard to the fact that the tanks may be designed to a proportion of either yield point or U.T.S., it is interesting to examine the design stresses in relation to mild steel physical properties. With a steel having a yield point of 15 tons/sq. in. and a U.T.S. of 28 the design stress becomes 10·5 tons/sq. in. based on the U.T.S. Such a stress is, of course, about equal in magnitude to a normal test stress, but, as is emphasised in the paper, the design conditions are also in the nature of a test.

Although there have been a number of suggestions for supporting the tanks to take account of contraction and expansion, and the various proposals must be treated as being confidential for commercial reasons, it is not giving away any secret to point out that, where there is a central fixed support, the tank supports at the periphery of a tank must all take account of the complete transverse contraction while those at varying fore and aft distances from the tank centre must take account of varying amounts of longitudinal contraction.

Regarding the impervious facing on the insulation, can the Author say whether this requires special properties and whether it is necessary for it to extend the full depth of the tank in the case of a design similar to Fig. 5 ?

If a design is developed with insulation on the inner side of the tank, then the inner surface of the insulation would also have to be impervious to the liquid and the insulation itself be strong enough for the pressure in the tank, but if a fracture occurred in this inner skin it might be very difficult to detect this, as no inert gas would be present. Could the Author comment on this?

Regarding the piping arrangements, I presume the material of the cargo piping must also be suitable for low temperature working, and it would appear that quite a lot of heat could be conducted along the piping or that very large temperature gradients could occur in the piping. Would this in fact be the case? In the case of a methane tanker, I presume the piping would require to be of aluminium alloy or do the pressure conditions render this unnecessary?

I note that it is contemplated to pump out the space surrounding the tank in the event of a fracture in the tank, and this piping also will require to have good low temperature characteristics and the associated pumps would also require to be safe. Is there any danger of the pumps not being able to function due to vaporisation of the cargo?

I think it would be most useful if a reproduction of the maximum degree of porosity in aluminium welds could be included in the paper.

Mr. G. H. Winders

This is indeed a most interesting paper and gives a good guide as to how to tackle the many varied and complex problems associated with a ship of this type. It is obvious that, due to the low temperatures at which these cargoes are carried, the insulation material is of the utmost importance and it is interesting to note that research is being carried out as to the possibility of fitting insulation inside the tanks. It is assumed that Balsa wood is the only insulating material which has been considered for the carriage of methane, and it is concluded that when the surface of the wood is suitably coated, then this is accepted as the "second barrier" and the internal structure of the ship may be of ordinary steel? The carriage of propane at $-40°$F. does not present so many difficulties, and no doubt it would be sufficient to insulate the tanks with a reasonable thickness of glass fibre. It is assumed that, as in the case of the *Methane Pioneer,* the cargo tanks will be built ashore and shipped in as separate units, and when considering the length of tanks, some consideration should be given to the problem of lifting the tanks in and out during classification surveys and at time of build, as any distortion whilst lifting will induce undesirable stresses in the structure.

It is regretted more information on the tank support system cannot be given at this time, but it would appear from the general description that the tanks will be supported on pedestals with anti-rolling and pitching chocks on the top and bottom of the tanks. In my opinion, side and end supports should also be fitted to restrain the tanks in the event of a collision, but the Author's comments would be appreciated.

As will be seen from the tank scantling calculations, it is most important to take account of the dynamic loading, but it would appear most unlikely that a tank will experience such a combination of rolling, pitching and heaving. No doubt some reduction in the scantlings of the tanks near midships can be allowed, since the dynamic loading due to pitching is in direct proportion to the distance of the tank from midships.

Mr. C. Dearden

My immediate interest as an engineer naturally centres on the pressure vessels, and I would like to have the Author's opinion on two points.

It is clear from formulæ printed in the appendices that, for low pressures at least, the wall thicknesses of the circular containers will be but a small fraction of the diameter and, having regard to the secondary stresses that may occur at any of the discontinuities due to anchoring of the vessel to hull, severe stress conditions could be expected in these areas.

In this connection, could the Author state what "g" could be anticipated in collision, and if the support devices are intended to plastically hinge under these conditions. Further, have any lobed pressure vessels been proposed for the carriage of

liquid gas as these seem to fulfil many of the requirements for this cargo.

The Author has presented a thoroughly interesting paper and is to be congratulated on the manner in which he has synthesised such a number of disparate facts and brought them together in one coherent whole.

Mr. A. G. KERSHAW

I would like to offer my congratulations to Mr. Davies for an extremely well presented and timely paper. As he so rightly states, there is a rapidly growing interest in the carriage of L.P.G. at low temperature—some 15 designs or projects have been examined in Ship Research during the past two years. It seems this interest was largely inspired by the publicity given to the successful voyages achieved by the *Methane Pioneer*—despite the fact that this experiment left the major problems unanswered. It must be remembered that the tanks in this ship never leaked, nor was she involved in a major collision or grounding. This happy state of affairs cannot last for ever, but until a major failure or accident occurs, there is no real means of assessing the effectiveness of the safety arrangements provided in these ships.

Still—in the absence of practical experience—requirements must be based on an assessment of the consequences of structural failure, and these are laid down in the recent addition to the Rules—Chapter D 70. Whilst these requirements are still comparatively new, I wonder if Mr. Davies would agree that, in some respects, they could now be reviewed. In particular, the section relating to tank design now appears a little artificial—especially the dynamic loading criterion. The calculations necessary to apply these to a ship's tank are somewhat tedious and it is a matter of conjecture if they are truly representative of actual service conditions. After all, tanks containing liquids have been built into ships for some considerable time without causing undue concern about the effects of rolling, pitching and heaving. I would suggest that L.P.G. tank design could well be based on the test head of water and the actual head of liquefied gas. The permissible stress for the test condition could be, as at present, three-quarters of the yield stress, and for the loaded condition, say three-eighths of the yield stress. In this latter condition credit could be given for the increase in yield and ultimate strengths that occur at the reduced temperatures. For steels suitable for propane tanks at $-50°$ F. the yield stress increases by about 25 per cent, and the ultimate stress by about 12 per cent. This should be borne in mind when considering the present seemingly high permissible stresses. Some form of dynamic loading must be kept to assess the suitability of the "keying" arrangements and a value of 1g vertically or horizontally would seem suitable.

The other section that is now open to question is the standard laid down for acceptance of tank material. The requirement for 40 ft./lb. Charpy at the service temperature for tank plate cannot be questioned, as the consequences of a brittle type failure could be disastrous. Difficulty has been found, however, in achieving the same standard in welds and the heat affected zone. It does seem that this requirement means the adoption of parent plate of a higher standard to keep the reduced properties of the weld zone above the 40 ft./lb. limit. Despite the high standard of weld quality that is required and 100 per cent radiography of tank seams, some notches and defects are bound to be present. Having laid down high standards of notch toughness to prevent cracks *propagating* in the tank plate, it would seem a lower level in the weld zone could be accepted.

There is no indication that a high notch toughness in the weld zone will prevent cracks *initiating*—protection against *propagation* into the adjacent plate seems much more necessary.

The presentation of this paper to the Staff Association has come at a very appropriate time. The first ship ever built solely for the carriage of liquefied propane or butane at low temperature sailed yesterday (March 5th) after completing the loading of its first cargo. Whilst this ship is not to the Society's class, I was given the opportunity of inspecting the arrangements and witnessing the preliminary low temperature trials. The design and safety arrangements provided on this ship are almost identical to those outlined in this paper, and it was heartening to find that the low temperature trials were a complete success. We can only hope that in the event of an accident the, as yet, untested arrangements will prove equally successful.

Mr. F. H. ATKINSON

I would like to thank the Author for his most informative paper on this comparatively new and intriguing subject, on which there are two or three points I would like the Author's opinion.

As the Author has said, a typical L.P.G. carrier will be one which has an L/D ratio of approximately ten and will sail at a relatively light draught. This will increase the vessel's susceptibility to slamming, and I wonder whether any increases are contemplated to the scantlings of the forward structure. Particularly when one remembers that the cargo is carried with a free surface and may be at a temperature of $-260°$ F. with all the inherent dangers of brittle fracture. Also, it should be remembered that the cargo tanks are not integral with the ship's structure and are only keyed in position. In this respect, I think Mr. Kershaw is wrong in advocating that dynamic stress calculations are superfluous, indeed I think they could be made more exacting in way of the forward cargo tanks. What I would suggest is that L.P.G. carriers be designed with fine lines forward, even at the expense of loss of cubic capacity, a substantial rise of floor and a ballasting arrangement which will give a draught forward of between ·025 and ·03L.

Turning to page 9, the Author suggests that the tanks should be sprayed during filling to prevent

or at least minimise thermal stresses. I would like to know whether he considers it necessary to repeat this operation when the tanks are discharging and similar conditions prevail, particularly when an inert gas is being pumped in.

With regard to safety, I would suggest, with all due respect to our electrical and engineering colleagues who are responsible for most of the safety precautions, a further device which is not dependent on any mechanical function for its efficiency, that is the addition of an odouriser to the cargo. This would not only prove efficient within the ship's structure, but also on the deck where gas could gather under an atmospheric temperature inversion when the tanks are boiling off at the mast head.

WRITTEN CONTRIBUTIONS

Mr. T. S. LEIGHTON (Nantes)

In view of the increasing interest and activity both in the U.K. and abroad in connection with the carriage of liquefied gases at low temperature, the presentation of this paper is most timely. The information it contains is of particular value to colleagues who have not had the opportunity to follow this comparatively new development.

I would like to contribute some remarks based on my recent experiences in surveys during construction of tanks for the carriage of liquid methane.

First, I would wholeheartedly endorse the Author's remarks on the care necessary to obtain an acceptable quality of welding, for what, at the present stage, must, it is assumed, be regarded as equal to Class I Pressure Vessel work.

The Provisional Rules require that all butt welds in these tanks be radiographed. This is a considerable undertaking, it being realised that in a methane carrier of the size quoted in the paper, the total length of welding to be so controlled can be measured in miles! The cost and the time required tempts the builder to adopt techniques which produce quicker, cheaper and consequently, less reliable radiographs.

In agreeing the standard of radiographic inspection, the Surveyor may find himself at variance with the constructor regarding the extent to which weld metal should be removed. It has been shown that the gain in time and cost from inadequate dressing of the weld may subsequently be lost with interest, as it leads to (a) radiographs which cannot be interpreted and (b) rejection of welds on radiographic evidence, because of defects above the acceptable limit which are located outside the plate thickness.

The Author refers to the acceptable limit of porosity in aluminium alloy welds, i.e. up to "scattered grade B", as described and illustrated in the B.S.R.A. Handbook H 3/58. Whilst this criterion offers some guidance to the Surveyor concerned, I believe it is intended for application to *reinforced* welds. Authorities in this field advise that a high level of porosity is indicative of an unsatisfactory technique likely to produce other more serious defects (i.e. lack of fusion), which frequently are not visible on a radiograph.

To achieve the required high standard of workmanship, the following are some of the precautions found to assist: —

Aluminium Alloy Tanks

Welding carried out as soon as possible after plate edge preparation, preferably on the same day. Where delay is unavoidable, the joint faces may be protected by being covered with adhesive tape, care being taken to avoid placing the adhesive on, or immediately adjacent to the prepared surfaces.

Final cleaning of the joint faces and plate edges by the welder immediately prior to welding, using a hardened steel scraper made to suit the angle of preparation.

Tack welds during assembly, made on the side opposite to that on which the first pass of weld metal is made.

Liquid dye penetrant not to be used, it being difficult to completely remove all traces of the dye which promote porosity in subsequent welding.

For satisfactory repairs, all defective metal to be removed. With defects in both sides of a double butt weld a successful repair is not possible by dealing with one side at a time. The cutting out of both sides before rewelding often results in the gap being so wide that it is preferable to remove a wide strip and insert a new piece of plate with two welds having correct joint preparations.

Experience has shown that it is highly probable some leaks will be found in the hydraulic test, especially where the design of the tank dome and cover attachments incorporate fillet welds which do not lend themselves to accurate radiographic examination. In view of the difficulty of ensuring complete removal of liquids for repair purposes, it is an advantage to carry out a preliminary tightness check using air or a suitable gas. Alternatively, a glass-fronted vacuum box with a soap solution on the outside of the weld may be used on flat or slightly curved shell plate welds.

9 per cent Nickel Steel Tanks

To achieve satisfactory welds, a special welding technique on the part of the operator is required; this technique is not easy to acquire, and a considerable number of welder approval tests, in all the positions to be used, are necessary before construction commences.

In the case of the tank surveyed by the Writer, the defects encountered mostly were porosity and very fine transverse fractures. Porosity was finally almost completely eliminated by baking the electrodes for about one hour at a temperature of 260° C. (500° F.) before use.

With transverse cracks, almost invariably located at the ends of weld runs and persistently in repair welds, the incidence was higher in the thinner plates and more especially in welds made in positions other than downhand. A high quality of radiograph is required to reveal these cracks.

The types of electrodes available for use with this steel have nickel contents varying between 60 per cent and 80 per cent. The correspondingly increased density of the deposited metal produces a marked variation in photographic density across the X-ray film. This variation is, of course, aggravated by failure to remove the excess metal. It might be mentioned also that the high nickel content in the weld precludes the use of magnetic flux inspection.

Passing on to the more general items in this paper, there are a few points on which I would appreciate the Author's views.

Regarding the flooding of spaces adjacent to the cargo tank containers in the event of a cold spot developing in the hull structure, will the Author confirm that arrangements for flooding only and not water circulation is intended for these spaces? In the former case, would he consider there was any serious risk of the insulation breakdown being of sufficient extent as to cause freezing and structural damage from water expansion in these spaces?

The Author states that, in the event of a fracture in the tank, the contents should be pumped overboard as quickly as possible. Would it be possible for Mr. Davies to enlarge on how this might be carried out?

Presumably, the discharge would be from nozzle extensions, or similar connections to the liquid cargo lines situated on the deck forward of the accommodation. Although, initially, complete vaporisation would occur at the point of discharge, it might be expected that local cooling of the atmosphere near the discharge stream would quickly permit the liquid to reach sea level, especially under static wind conditions. Expansion and vaporisation at sea level would seem to entail some risk of an explosive air/gas mixture reaching the upper works of the ship. Would the Author advise whether any recommendations are proposed in this connection? Perhaps regarding the number and position of discharge points.

On page 10, the Author states that pumping arrangements should be provided in the spaces surrounding the cargo tanks for the prompt removal of liquid gas which might leak from the tanks. This would seem an unnecessary requirement where the type of insulation fitted leaves very little free volume in this space. The Provisional Rules contain no reference to hydraulic or other form of tightness tests for liquid gas pipelines. I would appreciate the Author's comments on these points.

The complex liquid and gas piping systems, together with the large number and variety of associated valves, fittings, devices, etc., in these installations, requires the employment of a number of different wrought or cast materials. The suitability of several non-ferrous alloys and stainless steels for low temperature use being established, would it not now be of mutual advantage to all parties concerned if the Rules contained a list of acceptable materials, with their chemical compositions and temperature limits, in respective of which further special tests need not be carried out?

Regarding the periodical survey requirement that tanks must be internally examined annually, the warming up period necessary to permit entry into a liquid methane tank must be considerable, unless some means of heating is employed, either in the tank or in the Surveyor!

In conclusion, I would like to thank the Author for the useful information his instructive paper gives, on this extremely interesting development in marine construction.

Mr. T. E. DARBY

It is not often that the ship designer is faced with such a number of fundamental problems, and the fact that, in the evolution of L.P.G. tankers, these problems have been squarely faced instead of the usual reliance on an adaptation or gradual development of an existing type makes the reading of Mr. Davies' paper of more than usual interest. Although Mr. Davies has covered the ground very comprehensively, one or two points occurred to me on which I would like his comments.

Referring to Fig. 5, it would appear from the midship section that the inner bottom, shell and deck are intended to contribute towards the main longitudinal strength of the ship, although the inner bottom and deck are not shown continuous at the cofferdams in the profile. If this is so, the stress in some parts of the inner structure would be about 80 per cent of that in the outer structure and not sufficiently low to obviate the possibility of brittle fracture. This possibility is greatly increased by the danger of a cold spot developing, and under these circumstances, might it not be advisable to fit riveted seams in the inner skin as has been done in the outer shell and deck, in order to limit the extent of any fracture?

It would appear, at least in the arrangement shown in Fig. 5, that in the ballast condition the bending moment need not be very much greater than in the loaded condition. If the tanks formed by the double skin are flooded throughout the length of the cargo tanks (which might well be necessary for propeller immersion and prevention of slamming) the difference between this weight of water ballast and the cargo weight should not be very great in view of their respective densities. Is the possibility of having to flood a cofferdam or side tanks in the loaded condition taken into account in the longitudinal strength calculations?

On page 6, it is stated that thermo-couples must be fitted on the longitudinal and transverse bulkheads and on the inner bottom. Presumably, they are also required on the structure above the cargo tanks.

I note that arrangements must be made for the continuous sampling of the gas in the space surrounding the cargo tanks and for pumping out any liquid which may leak from the cargo tanks. What arrangements are to be made for the possible leakage of water from the side tanks, following (say) a collision or brittle fracture of the inner skin? The provision of sounding pipes to these spaces is complicated by their having to pass through the insulation and also by the presence of the pressurised inert gas.

Looking again at the midship section, it occurred to me that any of our colleagues allocated the task of ensuring that the insulation was adhering satisfactorily would need to be stout-hearted but definitely not stout-bodied.

I would like to thank Mr. Davies for a paper which is not only of great interest generally, but is also very valuable in explaining the purpose behind the requirements of the Society's Provisional Rules for Liquefied Gas Carriers.

Mr. N. S. Flensburg (Gothenburg)

Mr. Davies has explained the principles of the L.P.G.-carriers in an eminently clear way and his paper will be useful to anyone dealing with this new type of ship, and in my opinion can be considered to be one of the classic Staff Association Papers. I would like to ask the Author one or two questions.

1. It is not fully understood why the stresses in the ballast condition must always be checked. The amount of water ballast and its distribution in the longitudinal direction is about the same as for a normal bulk carrier. The weight of the machinery is only slightly heavier and the steel weight in way of the cargo tank range is no doubt heavier than that for a bulk carrier. Generally speaking, we could therefore expect the hogging stresses to be rather small.

2. In the case when methane is carried I have the feeling that even with very thick insulation fitted the actual temperature in the transverse double bulkheads will be rather much below zero and obviously brittle fracture possibilities would then be greater.

The temperature of the longitudinal bulkheads will, as far as can be seen, be higher than for the transverse sandwich bulkheads, however, even in this case the temperature will also probably be below zero when methane is carried and the risk for brittle fracture be present.

The conditions could be considered similar to those in the second and lower decks on refrigerated ships, when the decks are exposed to low temperatures, and it could be argued that a steel quality better than normal ship steel should be fitted in these bulkheads when methane is carried. I would also enquire if the Author knows of any temperature measurements which have been made for the transverse and longitudinal bulkheads when methane has been carried and what results, if any, were obtained.

3. On page 12 it is said that sufficient access space must be provided to allow room for removal of the insulation and a clearance is indicated between the cargo tanks and the insulation on the figure 5. This is heartily concurred in from the repair and special survey points of view. It is, however, known that methane carriers have been built recently without the access space and this system should in my opinion not be encouraged.

4. What is the reason behind the proposal that the permissible length of the individual cargo tanks should be about 50 ft. for methane and 90 ft. for propane? Should not the maximum length be related to the length of the ship as is now the case for ordinary tankers, further, should not the fact that the pressure on the bulkheads for methane is only about 70 per cent of the pressure when propane is carried, be taken into consideration? Would the answer be that the methane due to the low temperature is basically a more dangerous cargo and therefore a higher safety factor is desired?

AUTHOR'S REPLY

The Author is very grateful to all who have contributed to the discussion, particularly those who amplified certain important points.

Mr. Archer's contribution is most valuable in this respect and provides a useful background to the requirements for the carriage of gas under pressure which will shortly be incorporated in the Rules.

When considering the use of internal insulation it will certainly be necessary to show, by suitable tests, that the insulation is impervious to the liquid and will remain stable both under the temperature changes and under the dynamic action of the liquid.

Fig. 5 looks rather frightening and in this design deck and sheerstrake doublers would be relatively heavy for the size of ship. Generally speaking, in new construction there is no difficulty in obtaining the required longitudinal strength in view of the unusual depth of girder, but in conversions it will probably be necessary to fit doubling plates on the deck or sheerstrake.

The assumed conditions of roll, pitch and heave given in D 70 and used in Appendix II are undoubtedly severe, but must be read in association with the allowable stresses which are relatively high compared with those normally adopted in ship design.

Mr. Archer's last point is quite correct. Calculating the pressures at the mid-length greatly simplifies the work and only introduces a very small error.

Mr. Gray has both given some useful information and asked some pertinent questions.

It has not been possible or necessary yet to give a definite ruling as to the period for which the temporary containment must remain efficient. It is suggested that the maximum period which might be considered could be half the anticipated voyage and all materials so far proposed will certainly achieve this.

The amount of liquid to be left in the tank will depend on the length of the voyage and for a ten-day trip might have to be about 5 per cent of the tank volume.

The use of the boil-off as fuel is outside the Author's province, but he understands that the small amount of nitrogen involved is unlikely to necessitate special precautions.

Stripping arrangements would not normally be provided, since the residue will quickly boil off.

The Cleveland disaster is mentioned by Mr. Lockhart and this undoubtedly slowed down the widespread use of natural gas as a domestic fuel. However, it must be remembered that we now have a very much greater knowledge of the properties of materials at low temperatures.

Several speakers have drawn attention to the remark that the explosion risk with L.P.G. is no greater than with petroleum products. It was not the Author's intention to decry, in any way, the risks involved, and it might be as well to re-iterate that many petroleum products are extremely dangerous, and it is only by adhering strictly to the many safety precautions which have been developed over the years that these products are carried with the high degree of safety we have come to expect to-day. A similar degree of care will be necessary with L.P.G. cargoes.

Insulation in sheet or block form has been investigated on various occasions, but difficulty is experienced in maintaining vapour and liquid tightness at the junctions of the blocks.

It is confirmed that it may well be necessary to remove the tanks if repairs to the insulation are found necessary.

In his contribution Mr. Clayton has added much useful information regarding the pumping arrangements, and has corrected one or two statements in the original paper. The reason he gives for the abandonment of the proposal to use balsa wood as internal insulation is probably correct.

In reply to Mr. Sneddon's first point, the pressures quoted are those normally adopted by commercial operators. It is believed that most cargoes are a mixture and not 100 per cent propane so the pressure of 250 p.s.i. would probably correspond reasonably well with the temperature of 150° F.

In his second point Mr. Sneddon paints a lurid picture of the possible effects of a collision involving a gas tanker.

The Author is neither a physicist, chemist nor expert in heat transfer, so has felt it desirable to consult an independent expert in these subjects. He is assured that, while the effects of a collision would be serious, they would not be of the catastrophic nature envisaged by Mr. Sneddon. In support of this, the results were quoted of experiments in which liquid air was poured on to water. At the start there was a violent boil-off, but small saucer-shaped icebergs quickly formed. The liquid air was contained in these "saucers" and boiled off quietly. There was no large region of intense cold.

The expert added that, while he would not want to be on board either a methane carrier or a petroleum tanker in a collision, if he had to choose he would prefer to be on a methane carrier.

Like Mr. Boyd, the Author could have preferred to use the Centigrade notation, but felt he should conform to custom. The danger of water freezing if it should be necessary to flood the ballast space in an emergency is very real, and it would be necessary to keep the water in circulation.

Mr. Clemmetsen has provided some useful notes on the method of determining the hull scantlings.

In a design such as is shown in Fig. 5, the facing on the insulation should extend the full depth. With internal insulation, the insulation itself must be impervious to the liquid so that it would provide a barrier in the event of a failure of the facing.

It is confirmed that the piping would require to be of a material suitable for the temperatures involved.

It is regretted that it is not possible to provide an illustration of the radiograph.

It is confirmed that, as stated by Mr. Winders, if the surface of the balsa wood is suitably faced, then the internal structure of the ship may be of ordinary steel. Collision chocks are not considered necessary since, when the tanks are loaded, the friction between the tank and the bottom supports would be sufficient, together with the centre key.

Mr. Dearden has mentioned a most important point and there is no doubt that the number of attachments between a pressure vessel and the supporting structure must be kept to a minimum. The value of "g" to be anticipated during a collision is a very debatable point, since much will depend on the sizes and speeds of the vessels involved and the structural arrangements at the point of impact. Calculations have been made for varying masses, speeds and depths of penetration, and all that can be said is that a value in the region of 2g might be experienced if the structure was such as to provide high resistance to penetration.

Mr. Kershaw's work on L.P.G. carriers has been of the greatest assistance to the Author, but it is not thought that we can yet lighten his load by adopting his suggested tank design criteria. So long as so many widely varying tank arrangements are submitted, it is considered the fully comprehensive—even if somewhat artificial—dynamic criteria must be maintained.

The acceptance standards for the tank material and welds are under constant review, and it may well be that a slightly lower value than 40 ft./lb. for the weld zone would be acceptable as giving comparable properties to the parent plate. The results of many tests of differing types must be considered when determining an acceptable combination of tank material and welding process.

In reply to MR. ATKINSON, the conditions when discharging are unlikely to give rise to severe thermal stresses, since the change in temperature, and the difference in temperature over the depth of the tank only will be small.

The Author is most grateful to MR. LEIGHTON for sending a contribution, particularly as, at the time it was written, Mr. Leighton was in the throes of the completion of an experimental methane carrier. His remarks on the practices necessary to achieve the desired welding standard are most valuable and should be of great assistance in the future.

With regard to the general points raised by Mr. Leighton, it might be of interest that a very detailed heat-flow analysis has recently been carried out for a design of propane carrier where the insulation is not faced with a secondary barrier. The inner hull must therefore be of notch ductile steel and the detailed calculations show that it would be essential to maintain water circulation in the ballast spaces. A layer of ice would still form on the longitudinal bulkhead, but only of limited thickness. Water circulation is not, however, important when there is a secondary barrier.

Mr. Leighton's query regarding "dumping" is covered by a larger point raised by Mr. Sneddon. While the Provisional Rules contain no reference to hydraulic tests on the liquid gas lines, it would be considered good practice to test these to, say, 100 p.s.i. and to follow this with an air test at working pressure using soapy water at the joints.

Extensive testing is still proceeding on various materials, but it may, before long, be possible to consider Mr. Leighton's suggestion and prepare a list of materials suitable for various temperature conditions.

MR. DARBY is correct in assuming the inner hull as forming part of the continuous longitudinal material but, in view of the various precautionary measures already required, it has not been considered necessary to require riveted seams in the inner skin.

The load distribution, in all designs so far examined, has been such as to allow an ample margin should it be necessary to flood a side tank in an emergency.

Leakage from the side tanks would be a serious matter, since even a small amount of water entering the insulation and forming ice would cause a lot of trouble. Attention must be given to the arrangements on the longitudinal bulkhead so as to avoid hard spots.

In reply to the specific points raised by MR. FLENSBURG:—

1. While the stress in the ballast condition has not, up to now, been the governing factor for longitudinal strength, the hogging stress in this condition has always been greater than the sagging stress in the loaded condition. Trim and draught requirements appear to call for appreciable quantities of ballast in forward deep tanks and this, together with the uniform distribution of the ballast in way of the cargo tanks, causes appreciable hogging bending moments, so it has been prudent to check this condition.

2. It is agreed that in the transverse cofferdams the temperature would tend to gradually fall to something approaching that of the cargo space and, for this reason, these transverse cofferdam spaces should always be used as ballast tanks on the return voyage. The temperature of this space would then be raised to that of the sea water temperature. Measurements on the *Methane Pioneer* (the only methane carrier which has seen service) showed that the temperature of the hull steel work could be maintained at a safe level.

3. The question of the access space between the cargo tanks and the insulation has been the subject of much discussion. The Society cannot insist that sufficient space for access should be provided, but where this is not done, the Owners' attention has been drawn to the fact that partial removal of the insulation may become necessary and might cause difficulties. Some Owners appear to prefer the advantages of additional capacity and are willing to accept the disadvantage that at some time complete removal of the cargo tanks might be required.

4. The suggested maximum sizes of cargo tanks are based purely on practical conditions bearing in mind the contraction of the tanks on cooldown. The smaller figure for methane tanks is due to the much larger temperature difference (and hence contraction) and also because, so far, methane tanks have invariably been constructed of aluminium alloys, which have a much higher coefficient of thermal expansion than steel. It must also be remembered that the cargo tanks would generally be completed and tested before installation and this imposes a definite size limitation.

Various contributors have drawn attention to the desirability of breeding some dwarf surveyors able to work in cold conditions. With some of the designs which have been proposed, should any repairs be necessary, they will be very expensive, and this is a point which must be given serious consideration by designers and weighed against their desire to obtain the maximum possible cargo volume.

While several years experience has been obtained for the carriage of gases under pressure, the carriage at low temperature is still in its infancy. Whether or not this will develop depends primarily on political and economic considerations.

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